Comparative Analysis of Heat Exchangers and Continued Investigation of Variable Speed Versus Constant Speed Pump Systems in KU's Steam Power Plant

By

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Comparative Analysis of Heat Exchangers and Continued Investigation of Variable Speed Versus Constant Speed Pump Systems in KU's Steam Power Plant

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Abstract

The primary focus of this thesis is to investigate two particular heat exchangers in the steam power plant located at the University of Kansas. The secondary focus of this thesis is to compare the power consumption of two specific condensate pumps, also in the steam power plant located at the University of Kansas. The power plant generates and supplies steam (not electricity) to the buildings on the KU campus. The boilers in the plant require fuel (natural gas) to convert the feed water into steam. The two heat exchangers have been investigated to see how they affect the amount of natural gas that is required by the boiler during normal plant operation. Also, the power consumption of two specific condensate pumps, one constant speed and the other variable speed, has been compared during normal plant operation.

Heat exchangers have been widely used to substantially contribute to energy consumption savings, especially in power plants. There are presently two Bell & Gosset SU type heat exchangers in the steam power plant located on the University of Kansas campus. One of the heat exchangers is located in the basement of the power plant while the other heat exchanger (also known as the "vent condenser") is located on the first floor of the power plant. A comparative study has been conducted to investigate the amount of natural gas saved in one year by each of these heat exchangers.

There are two types of pumps that are used to supply condensate water in the power plant. One of these pumps is a Worthington D-824 constant speed pump, while the other is a pair of Grundfos CRE 15-3 variable speed pumps. These pumps provide water to the deaerator tank. The DA tank uses steam to preheat the condensate water and remove air and other non- condensable gases from that water before the water flows to the boilers. Both of these types of condensate pumps also move the condensate water up to the vent condenser on the first floor.

Different instruments were used for the measurement and recording of data in this thesis. The temperature data of the water was recorded by installing temperature sensors on the outside surfaces of the pipelines. These temperature sensors were installed at the inlets and outlets of the vent condenser and the basement heat exchanger. Temperature was recorded on a per-minute basis. Pump data such as discharge pressure, discharge flow rate and power consumption were also recorded on a per-minute basis. Pressure transducers were used to record pressure; magnetic flow meters were installed to record flow rate; and a power measuring device associated with the pumps helped in measuring power consumption. There was a common data acquisition system for all components. A Gateway laptop with HOBOWARE software installed was used to record and plot the data obtained from the respective data measuring instruments.

The basement heat exchanger's function is to heat the cold makeup water (i.e., the water received from the city of Lawrence) before it goes to the condensate storage tanks, which are also located in the basement. Also, this heat exchanger helps to reduce the boiler blowdown water temperature to a temperature less than 140^0 F (for environmental safety standards) before this water is drained to the sewers for disposal. In contrast, the vent condenser receives steam (along with associated non-condensable gases) from an open feed water heater in the basement (also called the Dearator tank or DA tank). This steam heats the condensate water flowing through the vent condenser. The vent condenser then returns this heated condensate water back to the condensate storage tanks located in the basement. Natural gas is used in the boiler to convert the condensate water into steam. The use of these two heat

exchangers saves natural gas which can be assigned a dollar value. It was found that the vent condenser saves approximately $27,680 for one year, while the basement heat exchanger saves approximately $7,660.

In continuation of the work done by a previous KU-ME graduate student, Raoof Alabdullah, the power consumption of both the Worthington pump and the Grundfos variable speed pumps was investigated for each month from June of 2015 through November of 2015. The average power consumption for the Worthington pump over this time period was found to be 5.29 kW. The Grundfos pumps can operate in either of two modes: pressure control or level control; and the average power consumption of the Grundfos pumps is considerably different, depending on operation mode. The pressure control mode of the Grundfos pumps is similar to that of the Worthington pump's operation, i.e., the Grundfos pumps run at a constant discharge pressure. Operation in level control mode depends on the water level of the DA tank. Based on how close this water level is to the target level (which is 52% of tank capacity), the Grundfos pumps either speed up or slow down, causing fluctuations in the discharge pressure. Also, while running in level control mode, the Grundfos pumps do not have enough pressure head to supply water to the vent condenser on the first floor. On October 21 and October 29, 2015, level control mode was employed from 1:30 pm to 3:30 pm; and the power consumption of the Grundfos pumps during this mode was compared to that of the pressure control mode runs made from 12 pm to 1:30 pm on those same days. The Grundfos pumps consumed 1.33 kW on October 21 and 3.66 kW on October 29, during the time periods when they ran in level control mode. This was considerably less than the 5.2 kW and 5.6 kW consumed by the pumps when they ran in pressure control mode on those same days. Thus, the level control mode allowed for significant decreases in power consumption.

A theoretical comparative study was also done between the Worthington pump and a less powerful condensate pump that would provide condensate water to the DA tank only. A Dayton constant speed pump was selected for this purpose. The vent condenser was assumed to be absent from the power plant, so no condensate water flowed to the first floor. This was compared to the current system with the Worthington constant speed pump supplying water to both the DA tank and the vent condenser. It was estimated that, for a 3% rate of interest, the presence of the vent condenser yields $20,296 in yearly savings as compared to just $2,470 if there were no vent condenser present and the pump were smaller. The savings calculations are with respect to the baseline: the Worthington pump being used without the vent condenser. The yearly costs of the Worthington pump and the Dayton pump were factored into the calculation of these savings, as was the cost of the vent condenser.

It is thus concluded that the vent condenser and the basement heat exchanger should continue to be used because of the large amount of natural gas savings. Also, the Grundfos pumps should be run in level control mode whenever possible, so that there is less power consumption by the pumps. In addition, it is recommended that the vent condenser be moved from the first floor to the basement near the DA tank, so that the Grundfos pumps can be operated in level control mode while the vent condenser is also in operation. By the simultaneous use of the vent condenser and the Grundfos pumps in level control mode, natural gas savings of an additional $15,000 for a period of 20 years might be achieved.

Different methods can be attempted in the future so as to improve calculation accuracy. To get even more accurate temperature data, instead of the external temperature sensors presently used, temperature sensors which are inserted into the pipelines could be installed at the inlets and outlets of the vent condenser and the basement heat exchanger. This would allow the sensors to be in direct contact with

the condensate water. The data acquisition system for such sensors would thus record the most accurate temperatures. Presently, only the flow rate of the condensate water for the vent condenser and makeup water flow rate for the basement heat exchanger are known. Flow meters could be installed in the shell side of both the vent condenser and the basement heat exchanger. The flow rate data of the condensed steam in the vent condenser's shell side and of the flash tank water in the basement heat exchanger's shell side would then be known. This would help to improve the accuracy of calculated natural gas savings. Also for future purposes, a flow meter with an associated data acquisition system could be installed in the makeup water line. This would help in recording the actual flow rate of makeup water, rather than estimating the makeup water flow rate based on temperature rise of the water as has been done in this thesis. Presently, there is no flow meter installed in the makeup water line. So, the makeup water's average flow rate is estimated based on the temperature rise in the makeup water across the heat exchanger and the hourly volumetric usage of water which is recorded "by hand" by the power plant staff.

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Nomenclature

A	= Contact surface area of heat exchanger (ft^2)
A_o	= Periodic yearly monetary value ($)
A_{calc}	= Calculated contact surface area of heat exchanger (ft^2)

bhp = Pump brake horsepower (hp)

BD = Blowdown flow rate (gpm)

c_p = Specific heat of water (Btu/lb_m°F)

C_1 = Units conversion factor ($1714 \frac{gpm\ lb_f}{in^2\ hp}$)

C_2 = Units conversion factor ($0.13368\ \frac{ft^3}{gallon}$)

C_3 = Units conversion factor ($\frac{1}{778}\ \frac{Btu}{lb_f - ft}$)

C_4 = Units conversion factor ($144 \frac{in^2}{ft^2}$)

C_5 = Units conversion factor ($60 \frac{min}{hr}$)

C_6 = Units conversion factor ($0.000293 \frac{kW-hr}{Btu}$)

C_7 = Units conversion factor ($0.00228 \frac{ft^3}{s\ gpm}$)

C_8 = Units conversion factor ($3600 \frac{s}{hr}$)

C_9 = Unit conversion factor ($86400\ \frac{s}{day}$)

d = Rate of interest as a fraction (-)

D = Diameter of heat exchanger tube (in)

E = Relative difference between power read from pump curve and HOBO data logger's recorded power (%)

E' = Relative difference between power calculated using Eqs. (9a) or (9b), and HOBO data logger's recorded power (%)

E'' = Relative difference between power read from pump curve and power calculated using Eqs. (9a) or (9b) (%)

E_{BFW} = Boiler feed water energy (Btu/hr)

E_{fuel} = Fuel energy (Btu/hr)

E_{steam} = Generated steam energy (Btu/hr)

$E_{vent/HEX}$ = Energy reclaimed by vent condenser or heat exchanger (Btu/hr)

$E_{vent/HEX,month}$ = Monthly energy reclaimed by vent condenser or heat exchanger (Btu/month)

f = Friction factor

FW = Feed water flow rate (gpm)

g = Sea level acceleration due to gravity (32.174 ft/s^2)

$h_{cond,in}$ = Condensate water enthalpy entering the vent condenser (Btu/lb$_m$)

$h_{cond,out}$ = Condensate water enthalpy exiting the vent condenser (Btu/lb$_m$)

h_f = Friction head (ft)

h_{minor} = Minor head losses (ft)

$h_{sat.steam}$ = Saturated steam enthalpy generated by boiler (Btu/lb$_m$)

$h_{sat.water\ (BFW)}$ = Boiler feed water enthalpy (Btu/lb$_m$)

H = Pump pressure head (ft)

L = External measured length of the heat exchanger (in)

LHV_{fuel} = Natural gas lower heating value (Btu/ft^3)

m = Flow rate of condensate water (lb$_m$/min)

m_{BFW} = Boiler feed water flow rate (lb$_m$/hr)

m_{makeup} = Flow rate of makeup water (lb$_m$/min)

$m_{makeup,hr}$ = Flow rate of makeup water (lb$_m$/hr)

m_{steam} = Generated steam flow rate (lb$_m$/hr)

$m_{steam,day}$ = Total steam flow in a day (lb$_m$/day)

$m_{vent/HEX}$ = Mass flow rate of excess condensate water or makeup water flowing to vent condenser or basement heat exchanger (lb$_m$/min)

n = Number of passes of heat exchanger (-)

ni = Expected replacement life of pump or vent condenser (years)

N = Number of tubes in heat exchanger (-)

P = Gauge pressure (psig)

PV = Present Value ($)

q = Heat transfer between fluids in heat exchanger or vent condenser (Btu/hr)

Q = Volumetric flow rate (gpm)

Q_T = Average volumetric flow rate (gpm)

$Q_{vent/HEX}$ = Volumetric flow rate of condensate water flowing to the vent condenser or basement heat exchanger (gpm)

S_G = Steam generated (lb_m)

t = Time (years, months, days, hr, min, s)

t_o, t_1, t_2, t_{j-1}, t_j = The 0th , 1st, 2nd, $j\text{-}1^{th}$, j^{th} time points for trapezoidal integration (s)

T = Temperature (^0F)

T_{c1} = Temperature of cold makeup water from water softeners (^0F)

T_{c2} = Temperature of heated makeup water going to condensate storage tanks (^0F)

T_{h1} = Temperature of hot water from flash tank (^0F)

T_{h2} = Temperature of water drained to the sewers (^0F)

Ti = Response time interval of the variable speed pumps' controller (s)

U = Overall heat transfer coefficient of heat exchanger (Btu/hr ft^2 ^0F)

V = Volume of natural gas (ft^3)

Vol_{fuel} = Volume flow rate of fuel converting BFW into steam in the boiler (ft^3/hr)

$Vol_{fuel,day}$ = Total daily volume flow rate of fuel used to convert BFW into steam in the boiler (ft^3/day)

$Vol_{fuel,saved}$ = Volume of fuel saved hourly by using vent condenser or heat exchanger (ft^3/hr)

$\dot{W}$ = Power consumption by pump as recorded by the HOBO data logger (kW)

$\dot{W}'$ = Work done to push condensate water to the DA tank (Btu/min)

$\dot{W}_1$ = Power consumption by pump as read from pump performance curve (kW)

$\dot{W}_2$ = Power consumption by pump, calculated from Eqs. (9a) or (9b) (kW)

$\dot{W}_{DA,\ vent\ present}$ = Work required to move water to DA tank and vent condenser (Btu/min)

$\dot{W}_{DA,\ vent\ absent}$ = Work required to move water to DA tank only (Btu/min)

$\dot{W}_{net}$ = Net work required to move water to vent condenser only (Btu/min)

Greek

γ = Specific weight of water (lb_f/ft^3)

γ' = Specific gravity (-)

ΔP = Differential pressure developed across a pump or pipeline (psig)

Δt = Time interval of recorded data (hr, min, s)

$\Delta T'$ = Temperature difference at the inlet of basement heat exchanger (^0F)

$\Delta T''$ = Temperature difference at the outlet of basement heat exchanger (^0F)

ΔT_{lm} = Log mean temperature difference (^0F)

ΔT_{rise} = Difference between exit and inlet water temperatures for the vent condenser or basement heat exchanger (^0F)

$\Delta T_{rise,max}$ = Maximum temperature difference between exit and inlet temperatures for the basement heat exchanger

η = Efficiency (-)

η_{boiler} = Boiler efficiency (-)

η_{HEX} = Heat exchanger efficiency (-)

η_{plant} = Plant efficiency (-)

ν = Specific volume of water (ft^3/lb_m)

ρ = Density of water (lb_m/ft^3)

Subscripts

1, 2 = Different pump states/speeds

A, B = Two different points along a length of pipe

BHEX = Basement Heat Exchanger

m = Motor

p	= Pump
v	= Variable frequency drive
VC	= Vent Condenser

Abbreviations

BFW	= Boiler Feed Water
BHEX	= Basement Heat Exchanger
CWP	= City Water Pump
DA	= Deaerator
DOE	= Department of Energy
HEX	= Heat Exchanger
LHV	= Lower Heating Value
MAWP	= Maximum Allowable Working Pressure
PM	= Plant Master
PV	= Process Variable
SP	= Set Point
TDS	= Total Dissolved Solids
VC	= Vent Condenser
VFD	= Variable Frequency Drive

Chapter 1: Overview and Details of KU Steam Power Plant

1.i Overview of Thesis

The project described in this thesis was developed for two purposes. The first purpose was to investigate the potential savings by the use of two specific heat exchangers in the steam power plant located on the University of Kansas' Lawrence campus. The two heat exchangers were the vent condenser on the first floor and the heat exchanger next to the flash tank in the basement of the power plant. The second purpose was to compare the power consumption of two condensate pumps, also in the steam power plant located on the University of Kansas' Lawrence campus. The two condensate pumps were the Worthington D-824 constant speed pump and a pair of Grundfos CRE 15-3 variable speed pumps, all located in the basement of the power plant.

Connected with the first purpose, the two heat exchangers had different functions. The vent condenser's main function was to heat up the excess condensate water, using the energy from the steam and non-condensable gases that was vented from the Deaerator tank (or DA tank), and then to return this water back to the two condensate storage tanks in the basement. In the basement of the power plant, softened city water or "makeup water" was transported to the same condensate storage tanks in the basement. Before the makeup water went to the storage tanks, it passed through a heat exchanger, also located in the basement. There was a flash tank located next to this heat exchanger, which supplied hot water to the heat exchanger. The basement heat exchanger increased the makeup water temperature, because of heat exchange with the hot water from the flash tank. This heated makeup water then went to the storage tanks. The condensate water (or feed water) that went to the boilers was thus at a higher temperature than it would have been without the two heat exchangers. Both of these heat exchangers helped in saving energy, because less energy was required to heat the boilers' feed water to form steam in the boilers. Therefore, less fuel was required, i.e., less natural gas was required, to burn for heating the boiler feed water.

Connected with the second purpose, pump data, such as flow rate, discharge pressure and power consumption, were also logged and analyzed as a continuation of Alabdullah's work [1]. From the analysis of this data, the benefits/drawbacks of constant vs. variable speed condensate pumps in the KU power plant were determined. The data in Alabdullah's thesis [1] only covered the months of February, March and April of 2015. Data from June to November of 2015 was recorded for this thesis; and the data was analyzed in order to see the effects of operation during different months (i.e., different weather conditions) on the pump comparison. May of 2015 was excluded because the power plant was shut down for 10 days for repairs and maintenance [which occurs each summer]. The boiler control systems in the power plant were also detailed. See Appendix A for information on the boiler control systems.

1.ii Steam and Water Circulation throughout the KU Campus

Most power plants have four major components – boilers, turbines, condensers and pumps. The KU power plant initially had all of these components. However, the turbines are not being used any more. The major reason for this is the cost of electricity production. For several years, the power plant was used as the

major source of electricity for the KU campus. However, in later years, it was found that electricity from the local provider (Westar Energy) was less expensive than that of the KU plant. For this reason, electricity production was discontinued; and the plant focused on providing steam to the campus, specifically for heating water and for HVAC systems. According to one of the power plant staff members, “the plant has not produced electricity since before I started working 30 years ago” [2].

Based on the location of the plant, the campus is divided into two distinct sections, namely the North campus and the South campus. The North campus consists of the buildings on the campus that are located north of the plant, while the South campus consists of the buildings south of the plant.

Steam that is produced in the steam drums of the boiler units has a pressure of around 175 psig. This pressure is then brought down to 90 psig (superheated condition) via regulator valves. The plant has two stations - the east station and the west station. Each of these stations has two regulator valves. Presently, the valves of the east station are being used to reduce the steam pressure (see Fig. 1), while the valves of the west station are not being used.

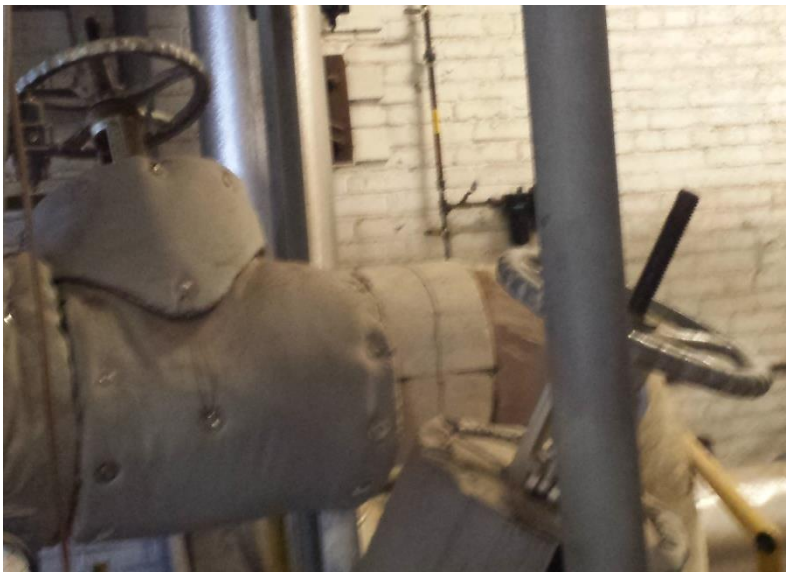


Fig. 1. East station isolation valves in the KU power plant reducing steam pressure from 175 psig to 90 psig

These two valves (for either the East station or the West station) are used alternately on a yearly basis, so as to maintain the lives of the valves. The reason there are two valves present in each station, instead of one valve, is to use the extra valve in case of failure of the other valve. Also, in case the steam demand exceeds normal capacity, both of these valves can be opened. The plant currently supplies 90 psig constant pressure steam to all buildings on campus. There are a total of three steam lines from the KU power plant to the buildings. There are two 14 inch pipes that send steam to the North campus and one 16 inch pipe that does the same for the South campus. The reason that there are two pipes for the North campus is because it has more buildings as compared to the South campus. Some of the steam at 175 psig, from the main steam lines, is directed to the Deaerator (DA) tank [3] **.** The DA tank (see Fig. 2) has a maximum allowable working pressure (MAWP) of approximately 50 psig. Thus, the 175 psig steam’s pressure is reduced to approximately 5-10 psig (superheated condition) so as not to over pressurize the DA tank.

The steam flow rate that is provided to the campus varies due to demand fluctuations. Production depends on many factors. For example, the demand in the Anschutz library and the Watson library is very inconsistent because of the libraries' humidity control units (i.e., summers are generally moist whereas winters are generally dry) [4]. Also, buildings like Mallott Hall have different requirements for steam, since people there use a lot of steam for cage washers and sterilization purposes [4]. Since the steam used in humidity control and sterilization is not recirculated back to the plant (i.e., this steam is essentially lost), the steam production can become irregular and makeup water is needed.

Also, the production of steam is much lower in the summer as compared to that produced in the winter. This is because the demand for hot water is much greater in the winter due to the cold temperatures. The greater the number of people, the more steam is required and vice versa. An interesting phenomenon is that the demand for steam increases considerably whenever there are KU basketball games in Allen Fieldhouse [4]. Since most of the games occur in Allen Fieldhouse from November to March (winter months), the opening and closing of the doors of Allen Fieldhouse by fans causes the steam demand to spike. Another factor affecting steam production is the number of people on campus. There is more steam demand when classes are in session as compared to the steam demand during summer and winter breaks. In addition, the KU Recreational Center uses a significant amount of hot water for showers in the locker rooms, which requires steam to produce [4].

Fig. 2. DA tank in the basement of the KU power plant

After usage at campus buildings, the steam condenses to water. By the time the water gets back to the power plant, its temperature is approximately 140^0 F. This water returns to the power plant through the North and South condensate return lines and then goes to the condensate storage tanks that are located in the basement of the plant. The mass of steam that is supplied to campus buildings is not the same as the mass of condensate water that returns to the power plant. There is approximately 30-40% steam loss (by volume), especially in Mallott hall, because of the cage washers and sterilization units. Also, there is continuous and periodic blowdown from the boilers (see Section 1.vi). Therefore, even though water is

recycled to the plant, there is a need for extra water. This extra water is known as “makeup water”, and the plant receives this water from the City of Lawrence. This makeup water comes to the plant at a pressure of approximately 25 psig (see Fig. 3). The plant has booster pumps, named “City Water Pumps” or CWPs, which increase the pressure of this makeup water to approximately 120-125 psig (see Fig. 4).

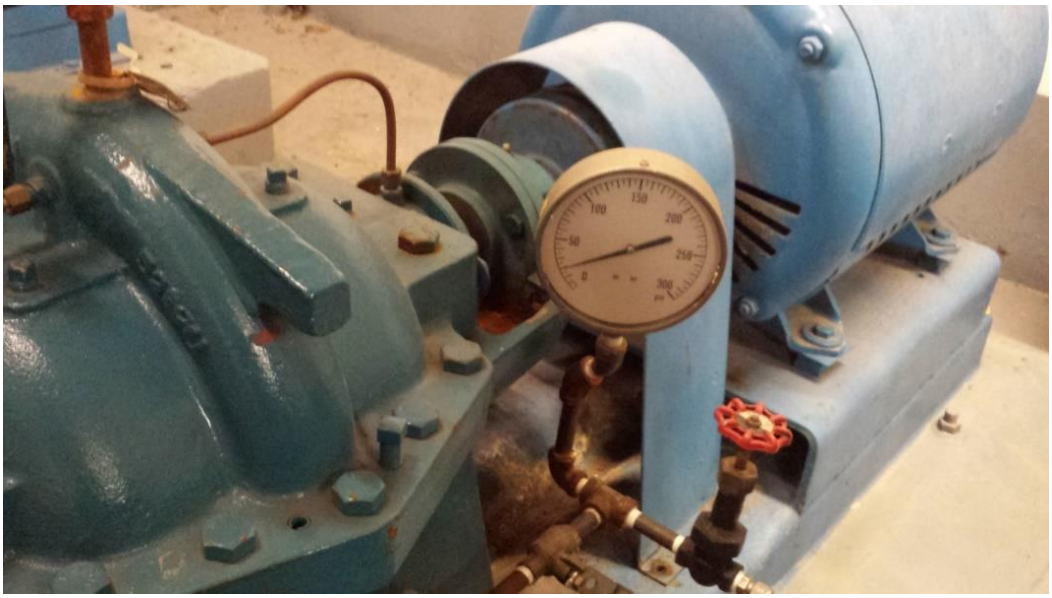


Fig. 3. Suction pressure of approximately 25 psig for CWP

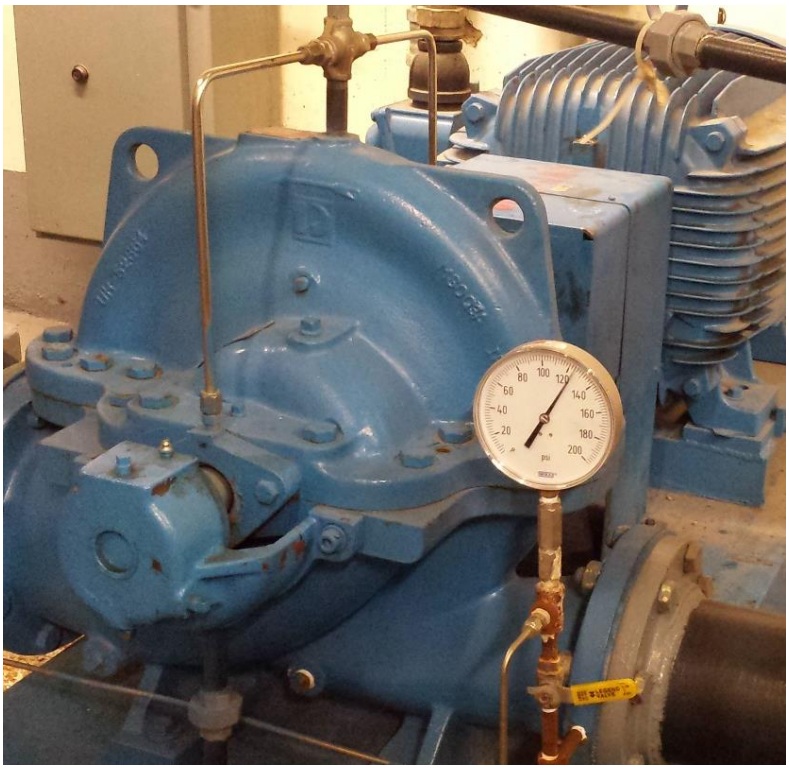


Fig. 4. Discharge pressure of approximately 122 psig for city water supply from CWP

The 120-125 psig water (boosted by the CWPs) is then circulated throughout the KU campus' centralized water system. The reason for increasing the pressure of the makeup water to 120-125 psig is because of the topography of the KU campus. Since there is an increased elevation of the campus from south to north, the 25 psig water does not have sufficient pressure to overcome this elevation (approximately 58 feet) and also overcome friction and fitting losses in the pipe. The plant then receives the 120-125 psig makeup water from the KU campus' centralized water system, which, after some heat addition and a pressure drop, then goes to the condensate storage tanks in the basement.

Contrary to what was originally thought by the author, the steam that the plant provides to the buildings is not condensed to supply hot water to the buildings. Instead, every building has heat exchangers which use this steam to heat up the colder building water.

1.iii Pumps

There are different types of pumps in the plant such as condensate pumps, boiler feed pumps and booster pumps. Each of these types of pumps has a different function. In the KU power plant, the condensate pumps move water at a set discharge pressure to the Deaerater tank (or "DA tank") and the vent condensing heat exchanger (or vent condenser). The function of the booster pumps is to boost the pressure of the water exiting the DA tank, to approximately 30 psig before it goes to the boiler feed pumps. The boiler feed pumps then increase the pressure even more so as to deliver the condensate water to the boilers located on the first floor. These pressures are usually on the order of 350 psig. Currently, there are four condensate pumps designated as pumps #1, #2, #3 and #4, all of which are located in the basement [5].

Pump #1 is a Worthington D-824 constant speed pump (see Fig. 5). Pump #2 is a pair of Grundfos CRE 15-3 pumps (see Fig. 6), running in parallel, both of which are variable speed. A schematic of the power plant is shown in Fig. 7a where pump #1 has been colored red while pump #2 has been colored blue. Pumps #3 and #4 are constant speed pumps. Pumps #3 and #4 have not been analyzed in this thesis, since there is no data acquisition system set up for them. If a single pump runs continuously for extended periods of time (more than one week), then overheating can occur which can compromise the viability of the pump and reliability of the power plant. Excessive vibration also can damage the mechanical seals of the pump. Due to these safety concerns, the power plant staff usually allow one pump to run each week, so as to keep all of the pumps in proper working condition. The job of these condensate pumps is to supply water to the DA tank in the basement and also to the vent condenser [6] (i.e., a heat recovery system) on the first floor, at an elevation of approximately 40 feet (i.e., 17.34 psig) above the basement floor level. See Figs. 7b and 7c. for the KU power plant's working model and a T-S diagram for the plant. The T-S diagram is explained in detail in Chapter 3.

The condensate pumps receive water from the storage tanks (see Fig. 8) (which are located close to the condensate pumps in the basement) and then discharge the water to the DA tank and the vent condenser. The discharge pressure is usually close to 43 psig. The variable speed pumps (termed pump #2) [1] can be operated in two modes – pressure control and level control. The power plant staff want to reclaim energy by using the vent condenser, so they always use the pressure control mode, as it allows the condensate water to go to both the DA tank and the vent condenser. However, operating in level control mode allows the condensate water to go to the DA tank but not to the vent condenser. This is because, during level

control mode, pump #2 has insufficient pressure to push the condensate water to the vent condenser on the first floor. Since the vent condenser is not used, there is no energy reclaimed when pump #2 runs in level control mode. The condensate water is delivered to the DA tank by means of a 4 inch diameter pipe, while it is transported to the vent condenser via a 2 inch diameter pipe [5]. The steam (along with non-condensable gases) that is released from the DA tank goes to the vent condenser. This steam is then used to heat the condensate water that the vent condenser receives during normal plant operation.

Fig. 5. Worthington D-824 constant speed pump (pump #1) in the KU power plant

Fig. 6. Grundfos CRE 15-3 variable speed pumps (pump #2) in the KU power plant

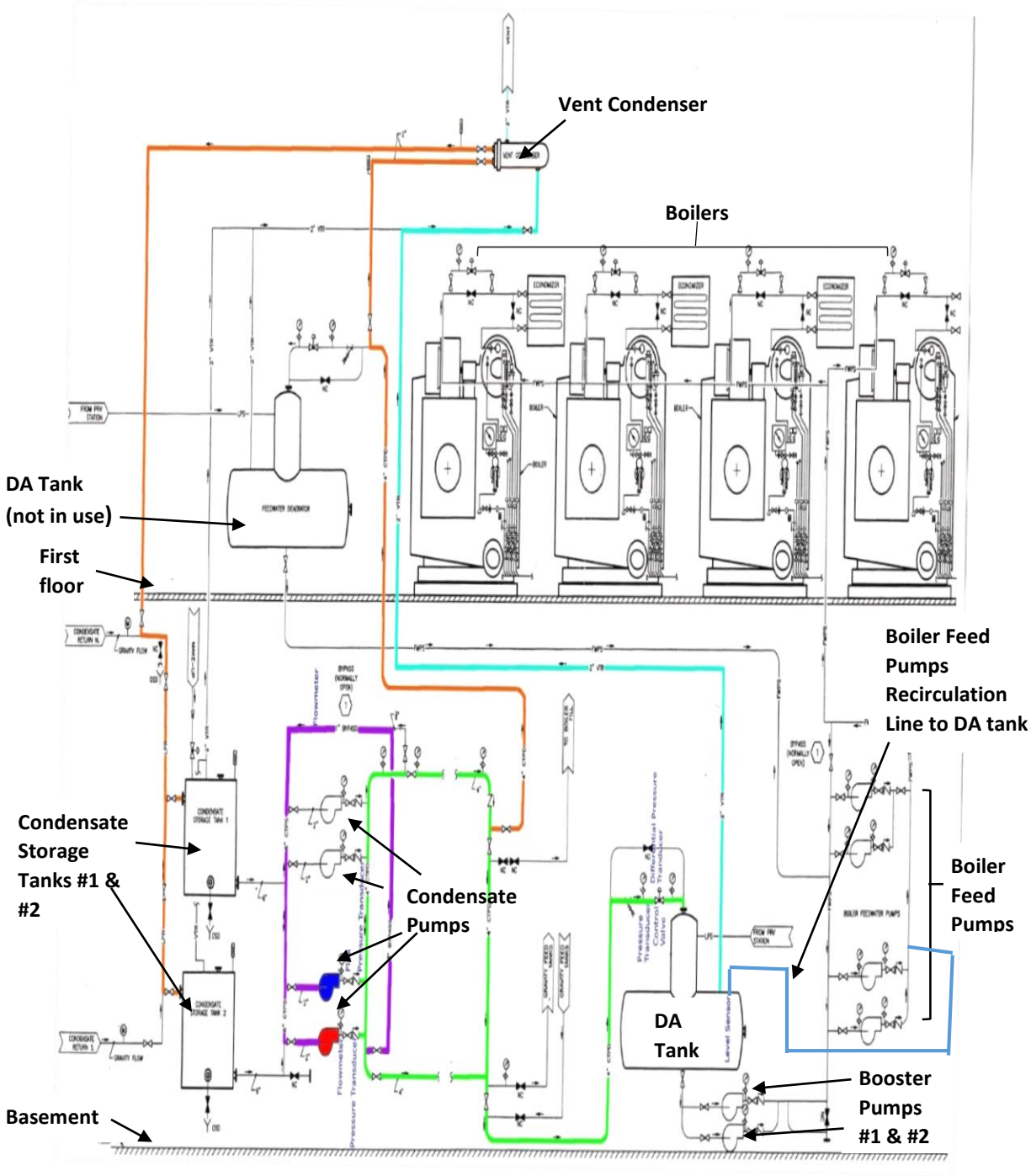


Fig. 7a. KU power plant schematic (Red Pump= Worthington D-824; Blue Pump= Grundfos CRE 15-3; Green line= Deaerator water supply; Light blue line= Non-condensables & steam line to vent condenser; Orange line= Condensate water supplied to vent condenser and back to condensate storage tanks; Purple line= Bypass to return water to pumps' suction side; Dark Blue Line= Recirculation line from boiler feed pumps to DA tank (reproduced from Ref. 1)

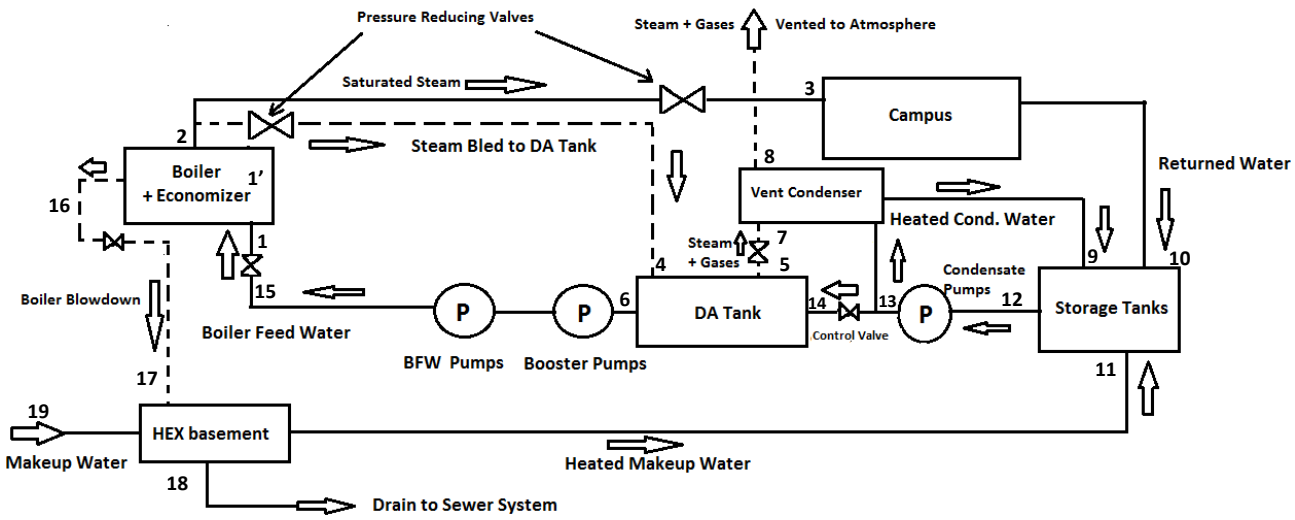


Fig. 7b. KU power plant working model

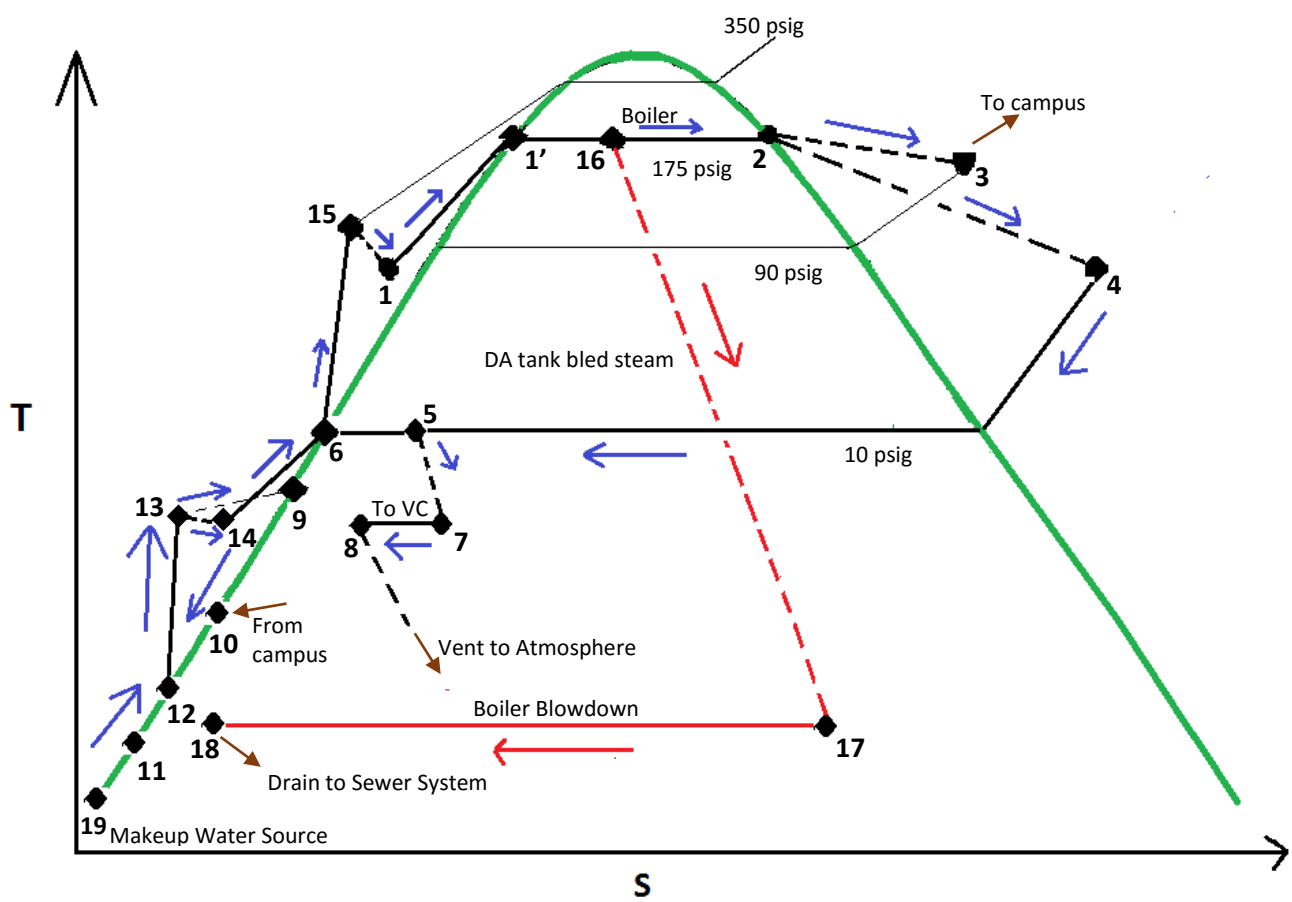


Fig. 7c. T-S diagram of the KU power plant (not to scale)

As previously stated, level control mode for the Grundfos pumps doesn't provide sufficient head (i.e., at least an extra 17.34 psig) for the water to go all of the way up to the vent condenser on the first floor. Discharge pressures of the Grundfos pumps vary from 5-25 psig, during level control mode. During the level control mode, since there is no condensate water flowing to the vent condenser, the steam (along with non-condensable gases) that comes out of the DA tank cannot be used to heat the non-existent excess condensate water. This steam and hot gases are then just vented to the atmosphere, thus being wasted instead of being used for energy recovery. During level control, the gate valve to the vent condenser was closed to prevent water flowing up to the vent condenser. This was not required, but, according to one of the members of the power plant staff, "we do not see any reason of keeping the valve open since condensate will not flow up anyway" [4].

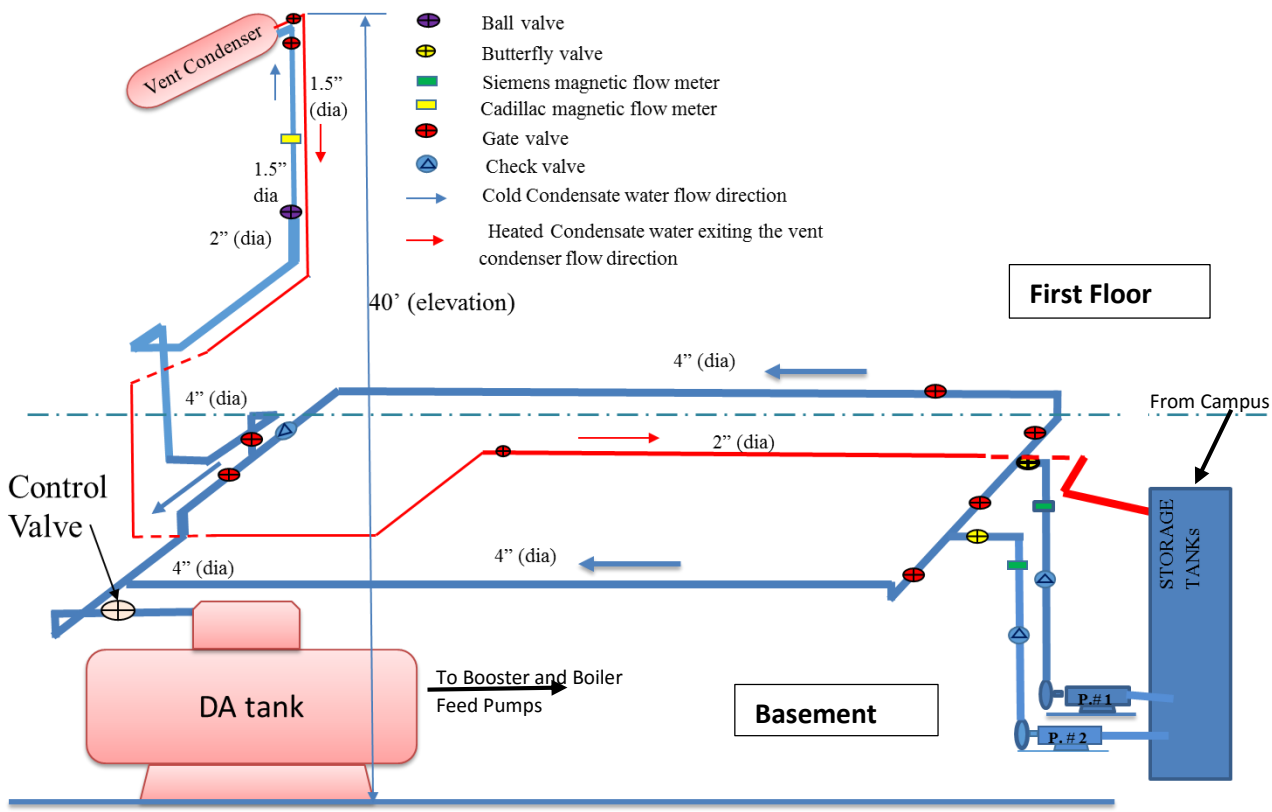


Fig. 8. Schematic of condensate water flow from condensate pumps to DA tank and vent condenser

The higher the temperature of the condensate water going into the boiler, the lower the amount of fuel required to heat up the water. This results in greater plant efficiency. Thus, there is a decrease in plant efficiency when the Grundfos pumps are operated in level control mode because the condensate water is not pre-heated. However, less pump power is required by the plant. Since there is no excess condensate water in the vent condenser during level control mode operation, the shell and tube material of the vent condenser gets overheated by the vented steam and non-condensable gases. This can result in damage of the vent condenser.

There is a pump recirculation line, or bypass line (see Fig. 7a, purple line) which is just downstream of the discharge point of the condensate pumps. Originally, according to Schmidt [9], this was thought to be the line that carried the excess condensate water back to the storage tanks. However, that was a misunderstanding because essentially all of the excess condensate water comes directly from the vent condenser on the first floor. The valve for this pump recirculation line is usually closed during normal plant operation. This valve is opened only when the gate valve to the vent condenser is closed so as to relieve the excess pressure in the discharge line of the condensate pump. This occurred when previous graduate student Alabdullah operated the Worthington pump without the vent condenser [1].

There is a control valve (refer to Fig. 8) that controls the amount of water flowing into the DA tank. However, there could be a situation where there is excess flow of water into the DA tank. This results in a higher water pressure on the inlet side of the control valve. Under these circumstances, the pressure has to be reduced; otherwise there is potential for water hammering in the pipelines which can seriously damage the pipes. Thus, the valve in the pump recirculation line is opened in this case.

Also, in the case of low steam demand, minimum flow of condensate water occurs. When a pump is run at low flow rate, the majority of the power input is converted to thermal energy. This causes the temperature of the water (in the pump casing) to rapidly increase, and some of this water can flash to steam. This steam can cause damage to the impellers, mechanical seals and other parts of the pump. In this case, for safety reasons, the pump recirculation line's valve is opened by the power plant staff to allow the water to be recycled to the storage tanks. There is also a recirculation line on the discharge side of the boiler feed pumps (see Fig. 7a (Dark Blue Line) and Fig. 9). This line goes to the DA tank and is only used for emergency purposes.

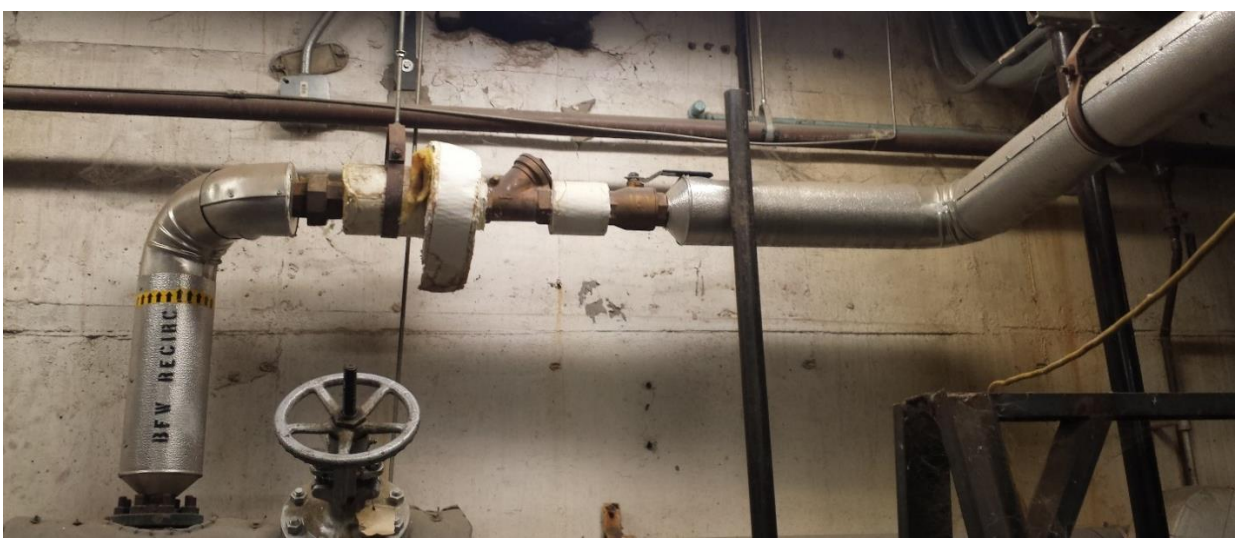


Fig. 9. One of the recirculation lines for the boiler feed water to flow to the DA tank in the basement

1.iv Deaerator Tank

There are two DA tanks (generally called open feed water heaters) in the power plant, one in the basement and the other on the first floor (refer to Fig. 7a). The one in the basement is a tray deaerator (see Fig. 10). The one on the first floor, which is a spray deaerator, is not in operation currently and has not been used

for the past three years. The reason the DA tank in the basement is currently in operation is because it has a higher capacity than the one on the first floor. The main purpose of a DA tank is to remove the entrained non-condensable gases (mainly oxygen and nitrogen) from the condensate water before this water is transported to the boilers. The remaining air and other non-condensable gases in the pipes have to be in very small concentrations (approximately 0.005 cm^3 of air per liter of water) so as to prevent corrosion of the pipes. To prevent air from leaking into the DA tank, it is usually pressurized at 5-10 psig. The DA tank has a MAWP of 50 psig. The condensate water, supplied from the condensate pumps, arrives at the DA tank with a pressure of approximately 35-40 psig. The control valve decreases the pressure of this water to 5-10 psig before it is delivered to the DA tank. The control valve maintains the level of water in the DA tank. Without the control valve, there could be excess flow of condensate water to the DA tank when there is low steam demand. The control valve helps prevent this situation by limiting the flow of condensate water into the DA tank. The desired level in the DA tank is 52% of the tank capacity by liquid volume.

In the DA tank, the water is heated by mixing with the small amount of steam that is bled from the total steam exiting the boiler. In a turbine-electricity-producing power plant, steam is bled from the turbines to the DA tank; but in the KU power plant, there are no turbines from which to bleed steam. The steam is essentially a heat source, having a temperature of approximately 370 ^{0}F. The DA tank, being tray type, has droplets of water falling through the trays to the water storage section [8] (see Fig. 10). These water droplets are heated and release a portion of the dissolved air. However, oxygen and nitrogen are not completely removed from the water by this process. So, the power plant staff also use chemicals, called oxygen scavengers (usually sodium sulphite) (see Fig. 11), to remove the remaining air.

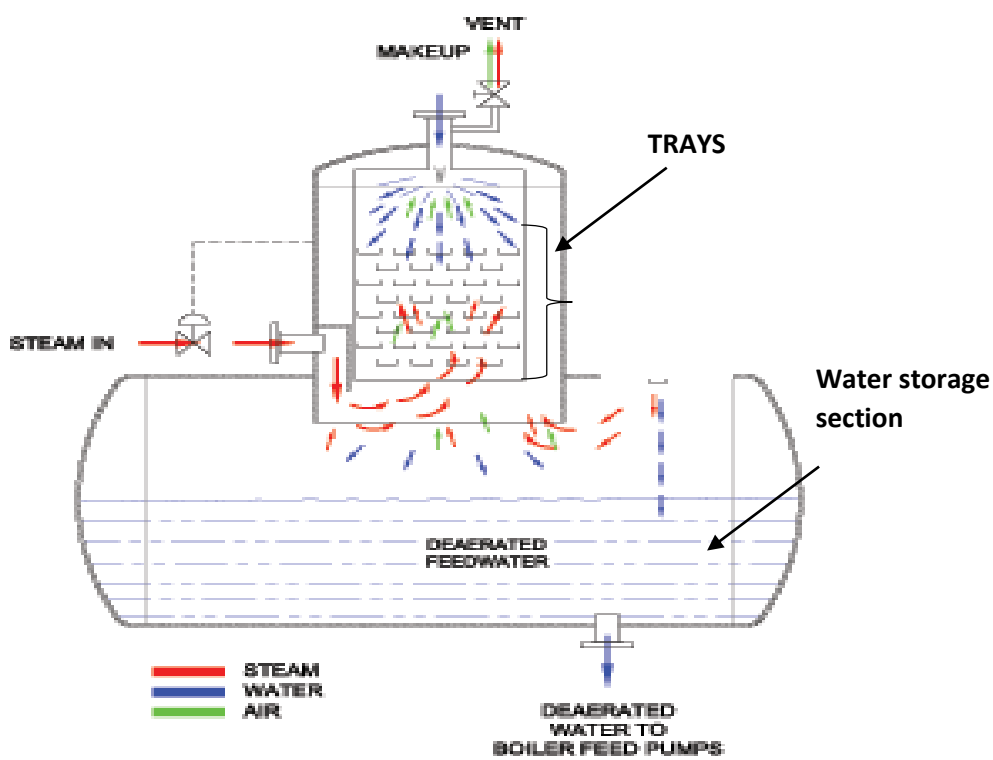


Fig. 10. Tray type deaerator (reproduced from Ref. 10)

Fig. 11. Tanks that store sodium sulphite solution before being pumped into the DA tank

There are two booster pumps located right below the DA tank (see Fig. 12). The power plant staff call the pump at the left "Booster Pump 1" and the one on the right "Booster Pump 2". The function of these pumps is to increase the pressure of the water that comes out of the DA tank (5-10 psig) to a pressure of 25-35 psig, before this water goes to the boiler feed pumps. Presently, only Booster Pump 2 is in operation. It is important to note that the water in the DA tank is very near its saturation point. Pumping this water can be dangerous for the piping, due to the potential for cavitation. To prevent cavitation, the elevation of the booster pumps is purposefully kept below the DA tank so that the pressure at the inlet of the booster pumps is more than the pumps' Net Positive Suction Head (NPSH) [9] (approximately 2-4 psig). In Fig. 12, it can be seen that Booster Pump 1 has an inlet pressure of 5-7 psig, even though this pump is not in operation. This pressure is due to the weight of the water in the DA tank.

Another role of the DA tank is the storage of water before it is discharged to the boiler feed pumps. Because an increase in feed water temperature leads directly to an increase in boiler efficiency [10], the DA tank is an essential component of the plant. It has been found that oxygen concentration decreases proportionally with temperature rise [11] (see Fig. 13). If the DA tank (which is at pressures of 5-10 psig) (see Figs. 7b and 7c) is operated at high temperatures, there is the possibility of water flashing to steam at temperatures of 100 ^{0}C and above. This steam-water combination can cause corrosion of the pipelines. Thus, the DA tank is operated at temperatures of 85-90 ^{0}C to prevent the formation of steam.

Once the oxygen and other non-condensable gases have been removed from the water, they are vented from the DA tank. The DA tank receives steam from two sources. One source of steam is the main steam line having a pressure of approximately 170-175 psig. This steam pressure is brought down to 5-10 psig via a regulator valve (see Figs. 7b and 7c). The other source of steam is a "flash tank" (see Fig. 14) located near the DA tank, that supplies some steam at 5-10 psig. The steam from the flash tank mixes with the steam from the main steam line, before steam is allowed to flow into the DA tank. (Details on the flash tank are provided in Section 1.vi.)

Fig. 12. Booster pump pressure gauges located underneath the DA tank

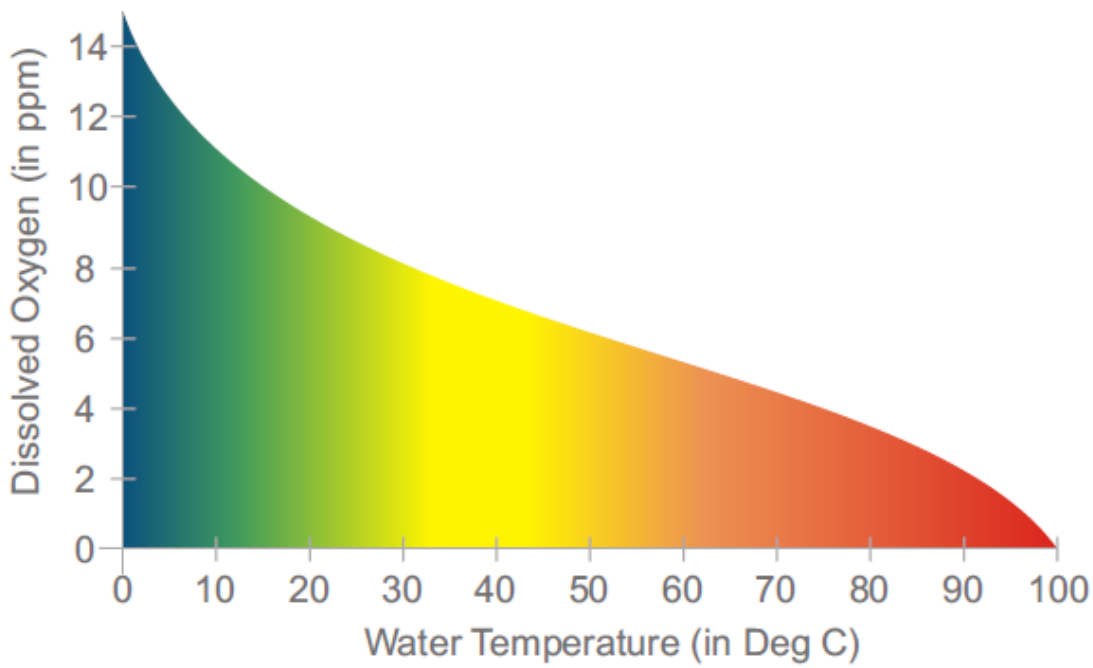


Fig. 13. Graphical representation of thermal deaeration at atmospheric pressure (reproduced from Ref. 1)

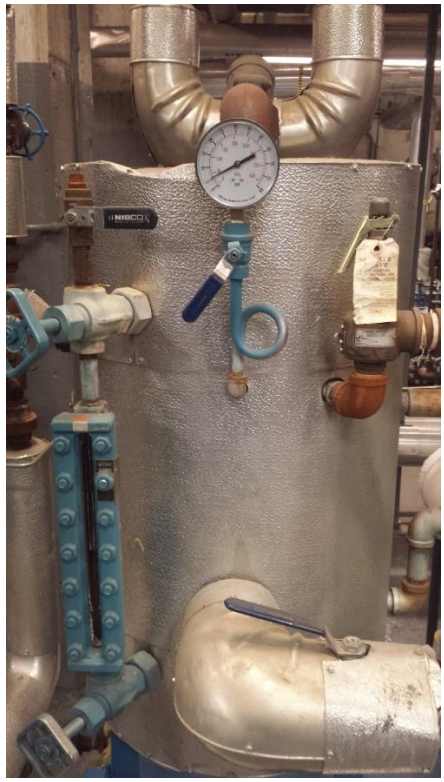


Fig. 14. Flash tank located in the basement of the power plant

1.v Vent Condenser

For many power plants, the non-condensable gases are vented to the atmosphere because these are not useful. However, there is a reasonable amount of thermal energy that these gases contain. To reclaim some of this energy, in the KU power plant, a vent condensing heat exchanger (or vent condenser) has been installed on the first floor, with the condenser's elevation being approximately 40 feet above the basement floor. The steam, with the associated non-condensable gases from the DA tank, is then directed to the vent condenser (refer to Fig. 7a and Fig. 8). The vent condenser is basically a shell-and-tube heat exchanger in which the steam condenses on the outsides of the tubes and the excess condensate water flows in the tubes [6] (see Fig. 15).

The temperature of the excess condensate water, flowing in the tubes, increases (by approximately 8 ^{0}F - 10 ^{0}F) because of heat exchange between the condensate water and the steam on the outsides of the tubes. This heated up condensate water then flows through 2 inch diameter pipes to the basement storage tanks, mainly under the influence of gravity (refer to Fig. 7a and Fig. 8). Some of the steam also condenses to liquid, but its "volume is negligible" as stated by the power plant staff [4]. There is no flow meter located near the vent condenser to measure this "negligible" water flow. This water also flows to the condensate storage tanks. The main benefit of having the vent condenser is that the condensate water is recycled while undergoing an increase in temperature. Boiler efficiency increases because the temperature of the feed water increases [10].

There are two condensate water storage tanks (see Fig. 7a and Fig. 16) located in the basement, near the inlets of the condensate pumps. Each of these storage tanks can hold close to 10000 gallons of water.

They typically store a little more than half of that capacity, approximately 6000 gallons of water each. This water is supplied to the condensate pumps. Then the pumps subsequently move the water to the DA tank and the vent condenser.

There are three sources of water for the storage tanks. The tanks receive water from the heat exchanger in the basement. Also, they receive the water returning from the campus pipelines of both the North and South campuses. In addition to the above two sources, the tanks receive water from the vent condenser on the first floor (explained in Section 1.vi). There is a valve located just before the inlet to each storage tank. These valves are usually completely open so that the tanks receive a continuous supply of water. These tanks then supply water to the condensate pumps.

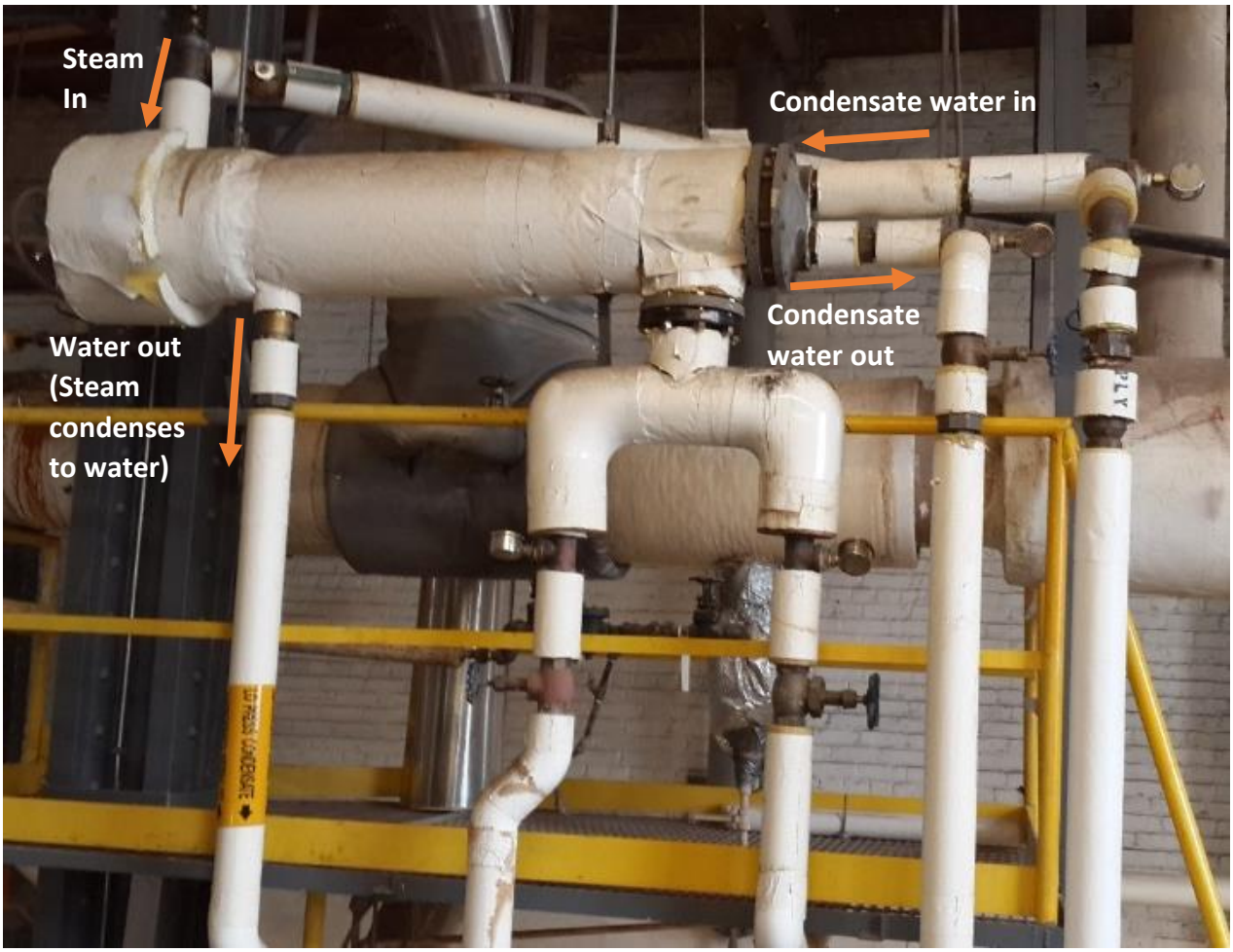


Fig. 15. Vent condenser on the first floor of the power plant

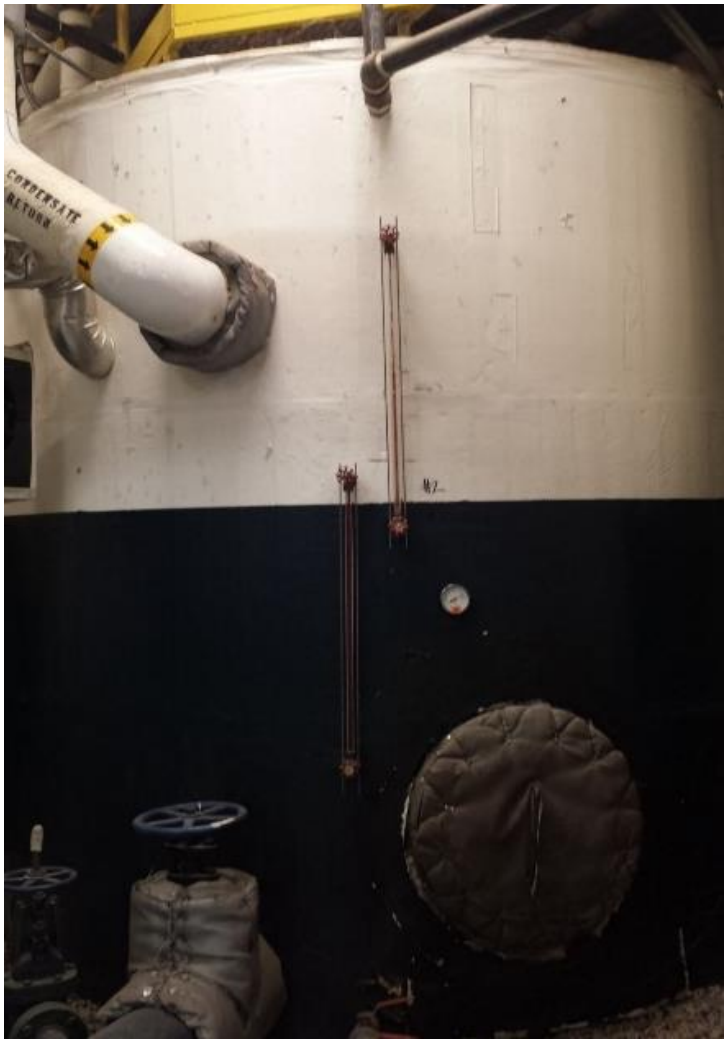


Fig. 16. One of the condensate water storage tanks in the basement of the power plant

1.vi Boiler Blowdown and the Heat Exchanger in the Basement

There are two types of boiler blowdown processes [12] that occur in the KU steam power plant: continuous blowdown and periodic blowdown. The level of TDS (total dissolved solids) has to be maintained at acceptable levels in the boiler feed water. For this reason, there is a continuous disposal of water, known as a continuous blowdown process. There is a water drum mounted low on the boiler called the “mud drum”. The function of the mud drum is to trap muddy water or “sludge” during circulation of the boiler feed water. The water in the mud drum is periodically disposed of every four hours or once every shift by the staff. This is known as a periodic blowdown process. This helps to remove the sludge from the boiler header, thus making the header clean. If this sludge is not removed, the sludge moves with the steam and thus pitting can occur, causing holes in the header.

Presently, the boilers are situated on the first floor of the power plant. Continuous blowdown [12] fluid goes to a steam flash tank that is located in the basement, near the DA tank. In the steam flash tank, the pressure is around 5-10 psig (the same pressure as the DA tank) with very high temperatures around 375

^{0}F. The boiling temperature of water at 5-10 psig ranges from 227^0 F to 240^0 F. Thus, upon reaching the flash tank, most of the water immediately flashes into superheated vapor, which gives rise to the name "Flash Tank" (see Fig. 14).

As explained earlier, the DA tank receives steam from the main steam line; this steam's pressure being reduced from 175 psig to 5-10 psig via regulator valves and becoming superheated. The steam produced by the flash tank mixes with the steam fed from the main steam line (once it has been regulated to a lower steam pressure, i.e., 5-10 psig). This steam mixture then enters the DA tank. The liquid water that remains in the flash tank is then directed to a heat exchanger (HEX) located close to the DA tank through 2 inch diameter pipes. This heat exchanger was initially thought by the author to be used solely for heating the cold makeup water from the water softening units to a higher temperature, before this makeup water went to the storage tanks, thus increasing boiler efficiency. However, it was later understood that its main purpose was to decrease the blowdown water temperature below 140 ^{0}F before dumping the water through a drain to city sewer lines, so as to stay within the city requirements of waste water temperature.

The basement HEX is a shell-and-tube type [13] and has two inlets and two outlets. One pair (inlet and outlet) is for the relatively colder makeup water, and the other pair is for the hot blowdown water from the flash tank. There is heat transfer occurring between these two fluids which has two effects: the makeup water is heated up before it goes to the storage tanks, and the blowdown water is cooled off before it is drained into the city sewer lines (see Fig. 17). The heat exchanger is explained in detail in Chapter 3.

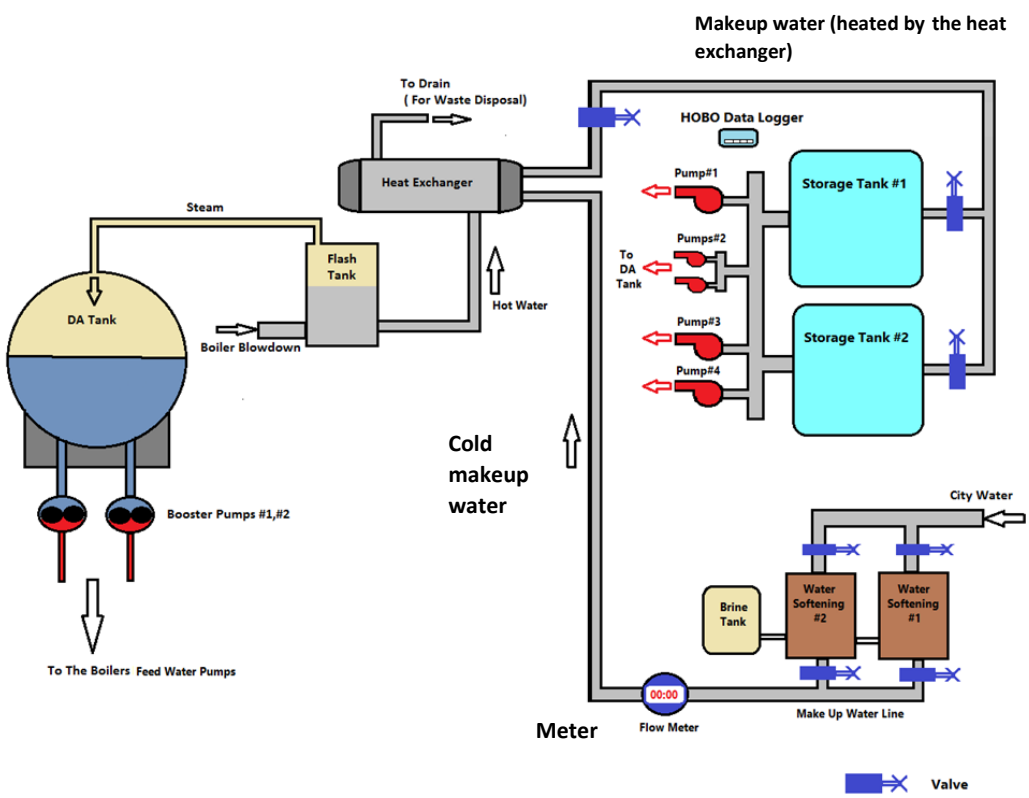


Fig. 17. Schematic of pathway of the makeup water from the water softening units to the storage tanks in the basement

1.vii Steam Drum

From the boiler feed water pumps, the feed water goes straight to the steam drums in the boiler units (refer to Fig. 7a and Fig. 18). There are tubes that are connected to each steam drum [5], through which there is flow of water, known as downcomers and risers. The risers are located in the furnace, which supplies heat constantly to the water. Due to this heat transfer, there is an increase in temperature of the water in the risers. Some of the water boils (approximately 60% of the water) and form bubbles, and wet steam (i.e., a steam-water mixture) is then formed. The downcomers allow the gravity-driven denser saturated liquid to go down to the mud drum at the bottom of the downcomers. This mud drum then settles out the impurities and solid particles from the water, and the remaining cleaner water then goes up the risers. The downward motion of the water in the downcomers pushes the steam-water mixture upward through the risers, until the mixture reaches the steam drum [5] (see Fig. 18). The difference in density between the water in the downcomers and the steam-water mixture in the risers, causes the upward motion in the risers. Thus there is no requirement for an extra pump to move the water up through the risers.

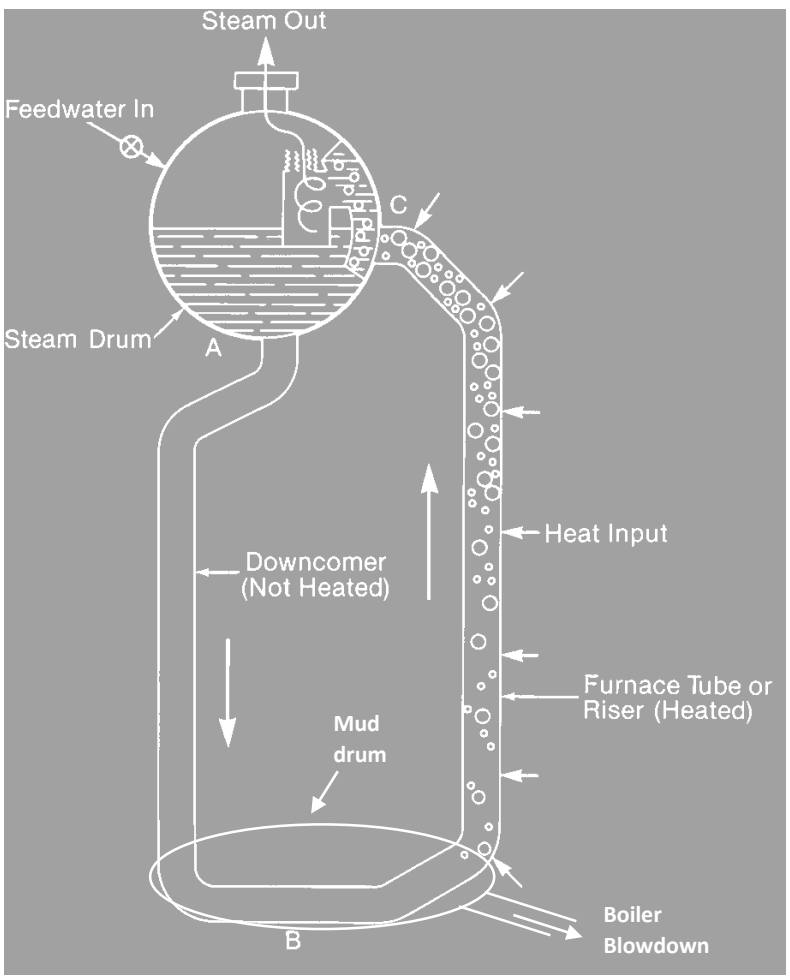


Fig. 18. Recirculation of water in the steam drum (reproduced from Ref. 5)

The steam drum of the boiler in operation has a fixed pressure of 175 psig. This 175 psig steam vapor is reduced to 90 psig via regulator valves, before the steam is sent out to the steam lines of the North and South parts of the campus. Usually, the steam drum is about half full of water. The sensor indicating the level of the water in the steam drum for Boiler #7 is shown in Fig. 19.

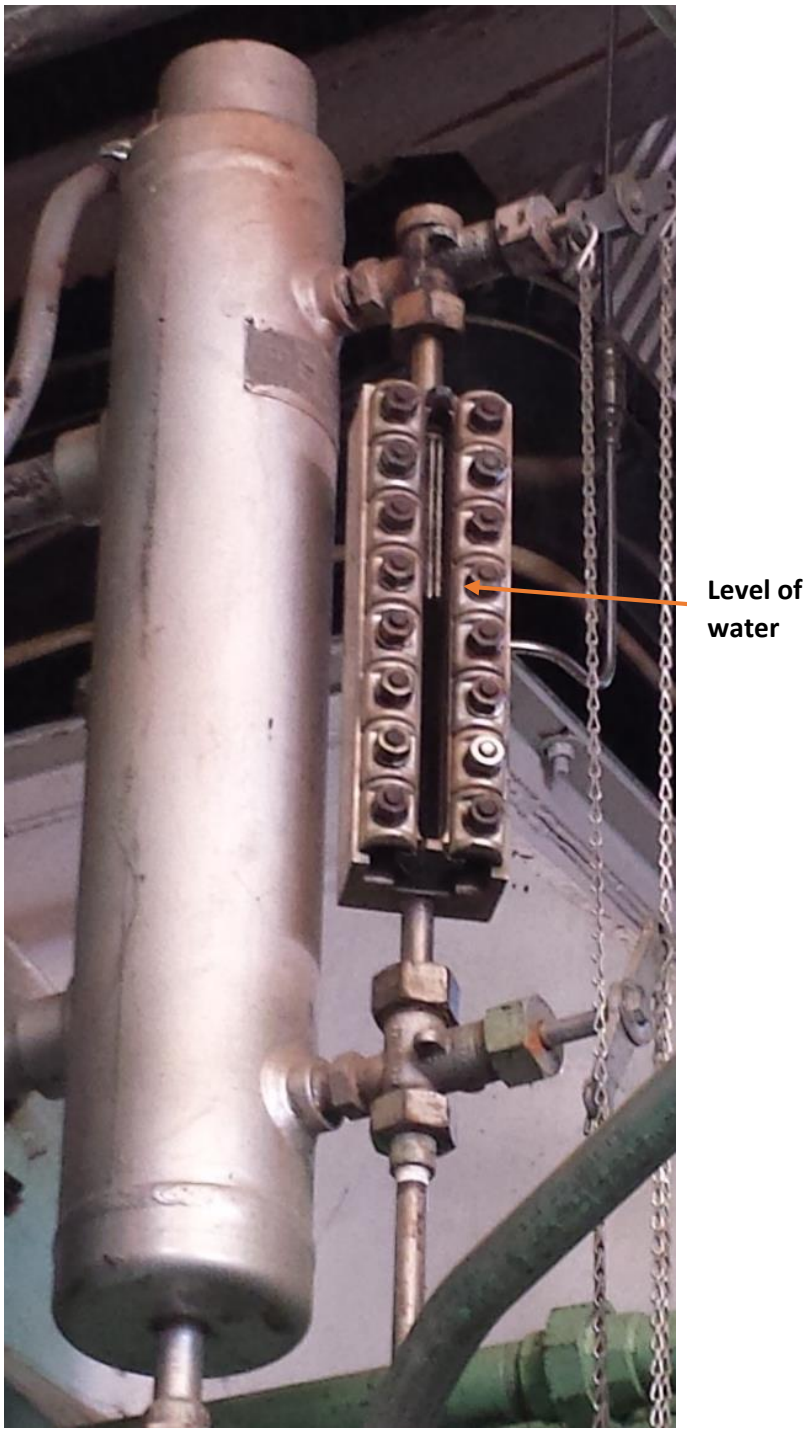


Fig. 19. Sensor indicating level of water in the steam drum for Boiler #7

1.viii Water Softening Units

City water has a substantial amount of dissolved calcium and magnesium, and is thus classified as “hard” water. Calcium and magnesium can severely corrode pipelines which carry water and thus have to be removed from the water [14]. Therefore, there is a water softening unit in the basement of the plant. This unit has two cylindrical tanks (A and B) that contain resin beds (see Fig. 20). The incoming city water flows through these resin beds. The resin beds attract and hold calcium and magnesium molecules that are in the water [14]. Only one of these resin beds is used at a time.

There is an electronic sensor (see Fig. 21) attached to the water softeners that continuously determines the amount (in multiples of 100 of gallons) of the resin bed that is free from calcium and magnesium. This sensor counts down from a maximum of 11,000 gallons to a minimum of zero gallons. Once the sensor value reaches zero, the resin bed tank stops operating. Control then shifts to the alternate resin bed tank, which then starts operating.

When one of the resin bed tanks is completely contaminated with magnesium and calcium, it needs to be purified. The water softening unit also has a large tank (known as the brine tank) which contains sodium chloride. City water flows into this tank to form a solution of sodium chloride or brine (see Fig. 22). The brine water is pumped into the resin beds. The brine is used to purify the resin beds and remove all of the magnesium and calcium molecules. The contaminated solution of brine (containing dissolved calcium and magnesium) is then drained into the city’s sewer system.

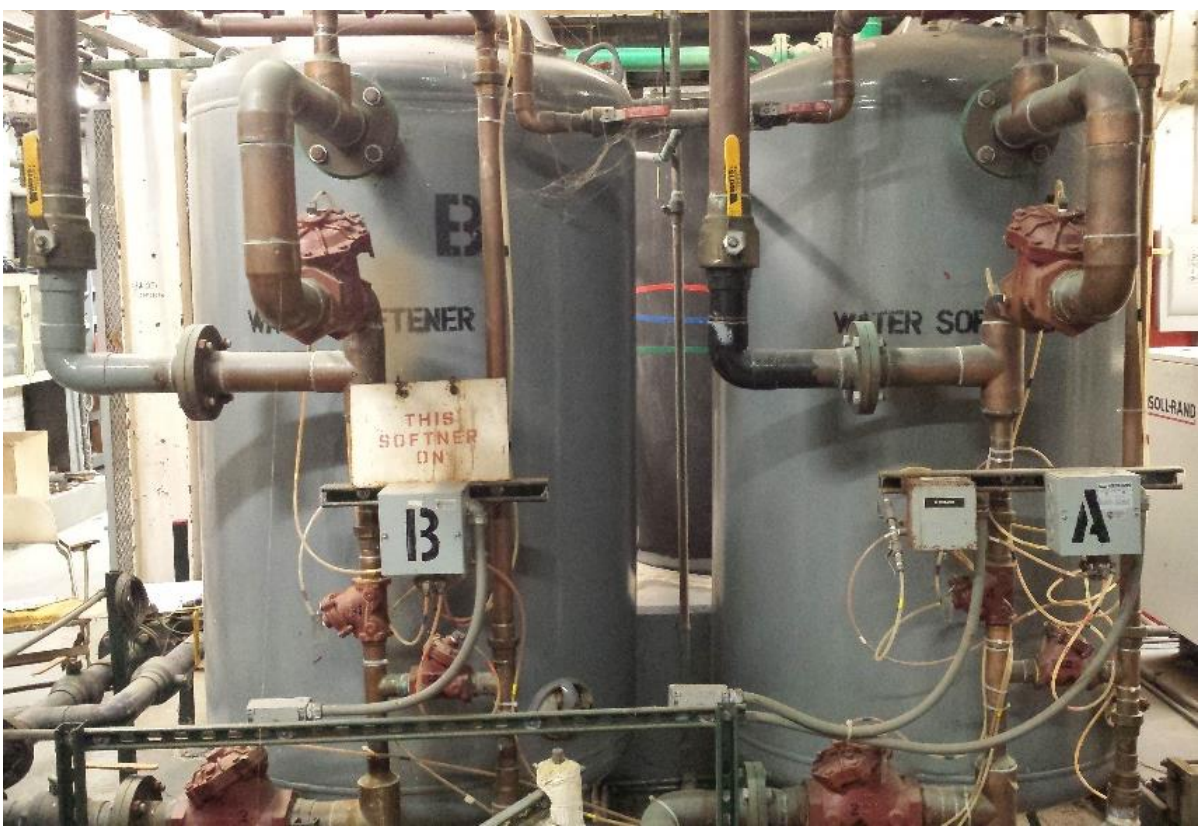


Fig. 20. Water softening units containing resin beds

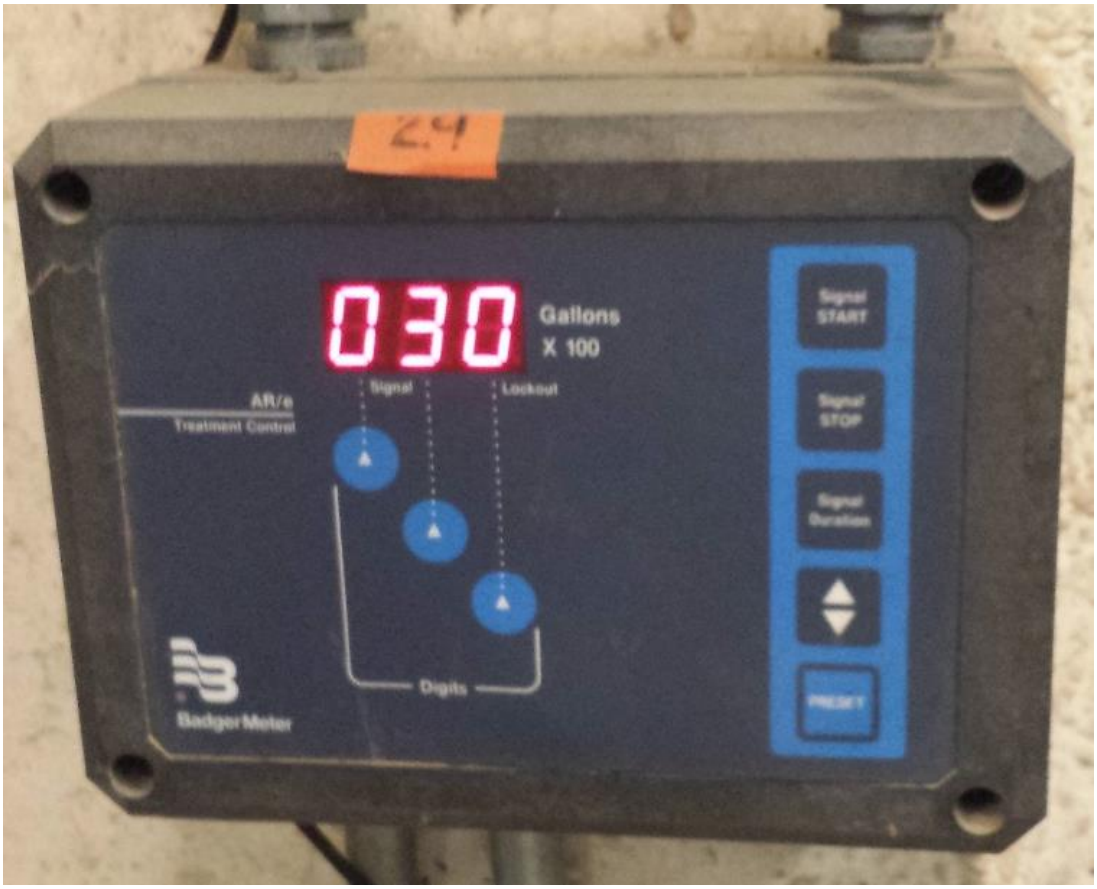


Fig. 21. Electronic display for water softening unit

Fig. 22. Brine tank next to the cylinders containing resin beds

1.ix Air Compressors

The power plant also has a pair of air compressors in the basement that supply high pressure air to the machines that require compressed air for operation (see Fig. 23). Most of this air is used to operate the control valves in the plant. The air compressors produce air at a constant pressure of 113 psig, and provide cooling air to the flame scanners present in the boiler. The flame scanners monitor the fires produced in the boiler and detect the presence or absence of flame in specific regions of the boiler.

Fig. 23. One of the two air compressors in the basement of the KU power plant

1.x Fuel

The power plant uses natural gas as its principal fuel. The source of this natural gas is the Black Hills Energy company of Kansas. The natural gas is supplied to the KU campus via pipelines at a pressure of 25 psig, with the main hub being near Naismith Hall. From this main hub, a pipeline goes to the KU power plant (see Fig. 24) at that same 25 psig pressure. Before the gas enters the power plant, the pressure is regulated by a control valve and is reduced to 17 psig. The natural gas is then piped to the boilers of the power plant. Each of the boilers in the plant has a different input natural gas pressure, regulated by individual control valves. Boiler #8 takes in the natural gas at 17 psig (see Fig. 25), and Boiler #7 reduces the pressure of the natural gas to 16 psig. The older Boilers (#1 and #2) reduce the natural gas pressure to 10 psig.

During emergencies, when natural gas is not being used as the fuel, the power plant uses diesel as a substitute. Diesel has its own piping, pumping and valving systems, which are located in the basement of the power plant. The diesel fuel is stored in large tanks outside of the plant (see Fig. 26). When diesel fuel is used, all of the natural gas valves are closed in order to allow for flow of the diesel into the main plant. This diesel runs to the same boilers, however in separate pipelines and injection systems.

Fig. 24. Natural gas pipeline just outside the power plant

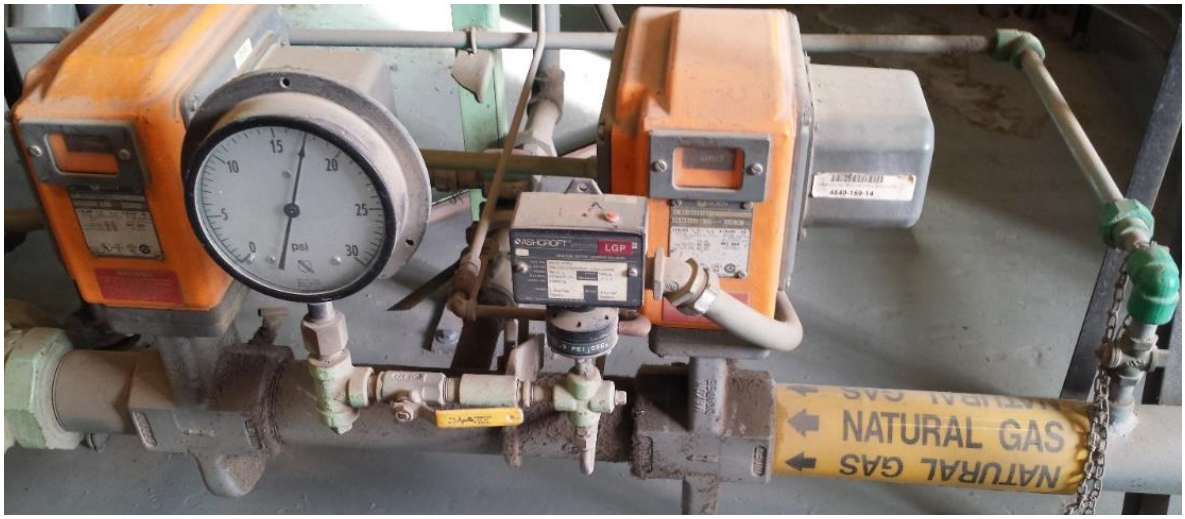


Fig. 25. Safety shutoff valve with pressure of 17 psig for Boiler #8

Fig. 26. One of the diesel tanks outside the KU power plant

Now that the overall power plant features have been covered in detail, Chapter 2 covers the various instruments and data acquisition systems that have been installed in the power plant to collect the data that were used for this thesis. Chapter 3 covers the calculations of energy saved by the two heat exchangers in the power plant. Chapter 4 covers the data obtained for the Worthington and the Grundfos pumps and the differences in power consumption found when the Grundfos pumps were run in pressure control versus level control modes. Chapter 5 provides the conclusions obtained from this project and also the recommendations that can be made for future work. Appendix A explains the boiler control systems that are being used for boiler operation. Appendix B covers the measuring devices that have been used in this project. Appendix C covers the steps that have been taken for the calibration of the temperature sensors used in this project. Appendix D explains the Linear Scaling Assistant Window of the HOBOWARE software used for recording pump data. Appendices E and F cover the calculations that have been done for the boiler efficiency and the heat exchanger energy gain, respectively. Appendix G covers the pump curves for the Worthington and the Grundfos pumps along with specifications. Appendix H covers the plots of discharge pressure and discharge flow rate of the Worthington and Grundfos pumps from June of 2015 to November of 2015.

Chapter 2: Instruments Installed and Data Logging Equipment

This chapter covers the measuring devices and instruments that have been installed in the KU steam power plant. Most of the measuring devices were installed by Schmidt [7] with some installed by Alabdullah [1]. These devices have continued to be used for recording data. Additional devices have been incorporated by this author in order to obtain temperature data for the vent condenser and the basement heat exchanger. The devices used are summarized in Table B1.

Also, data logging procedures and the subsequent analysis of the data, with respect to both heat exchangers and the Worthington and the Grundfos pumps, are covered briefly in this chapter. Both pressure control and level control modes of operation are described for the Grundfos pumps.

2.i System Setup

This section covers the temperature sensors, flow meters, pressure transducers, power monitoring supply, and the level control system that have been installed in the KU power plant. The locations of these installed devices are described in this section. To make sure that the data recorded by these devices were accurate, each of these instruments was calibrated. In certain cases where the devices were found to be faulty, the devices were replaced.

2.i.a Temperature Sensors

In order to measure the temperatures of the water at the inlets and outlets of both of the heat exchangers, i.e., the one in the basement and the vent condenser on the first floor, analog temperature gauges were initially used [8]. However, the temperatures from the gauges could not be trusted because the temperatures were not recorded digitally and accuracy was limited. So the only way to get these temperatures was to view the gauges carefully and to trust the judgement of the viewer. Also, even if the viewer's judgement could be trusted, the temperature was recorded only for the particular period of time (i.e., 5-10 minutes) that the viewer looked at the gauge, and not for a longer time interval, such as a day or a week. The least count of the gauges was 1^0 F. The gauges were old and hadn't been calibrated for a number of years.

For these reasons, digital temperature sensors were installed that could accurately record the temperatures of the water flowing in the pipelines at the inlets and the outlets of the heat exchangers. Ideally, a temperature sensor should be installed inside the pipeline. However, that was not allowed by the power plant staff, as that would require cutting holes in the pipe for the sensors to be installed, which meant that the power plant could not operate during installation. The power plant is down for maintenance for only one to two weeks in May of each year, and this project started in June of 2015. Also, such an installation would be expensive.

Hence, temperature sensors were installed on the outsides of the pipes. After much investigation, the HOBO TMC6-HE temperature sensor was selected (see Appendix B1). A thermal paste was applied to these sensors (see Fig. 27) for optimal contact with the pipe, and the entire probe and pipe were insulated so that outside temperatures would have minimal effect on the measurements.

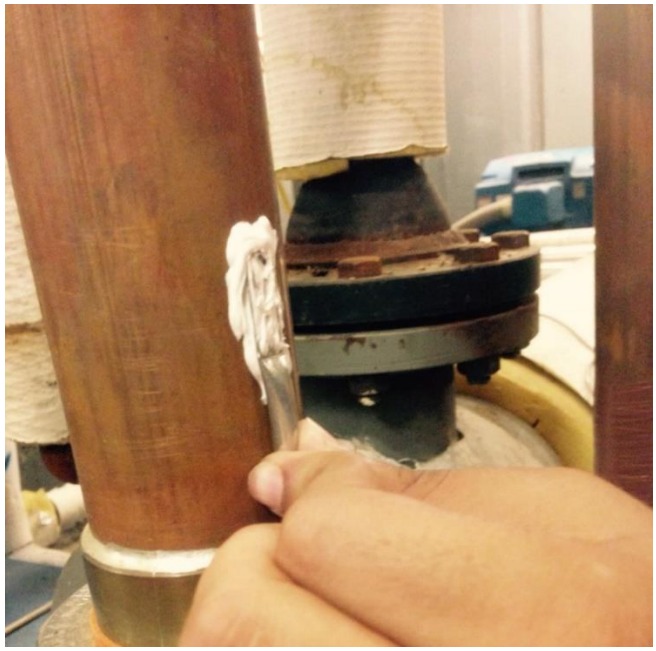


Fig. 27. Thermal paste applied on the pipe surface to enhance conduction to the temperature sensor

A plot of the temperature sensor's error, given in Fig. 28, shows the accuracy from 0 ^{0}C to 100 ^{0}C (i.e., 32 ^{0}F to 212 ^{0}F). The range of measurement at the power plant for the basement heat exchanger was 0 ^{0}F-240 ^{0}F and for the vent condenser was 130 ^{0}F-180 ^{0}F. Based on this plot, it could be assumed from linear extrapolation that the sensors would have approximate errors of 1 ^{0}F to 2 ^{0}F for temperatures from 212 ^{0}F to 240 ^{0}F. For the vent condenser's temperature range, the error of 0.36 ^{0}F to 0.72 ^{0}F was not ideal, since the temperature rise of the condensate water as it flowed through the vent condenser was of the order of 10 ^{0}F. However, since these sensors were compatible to the HOBO data acquisition software that was already being used, they were used for this project. The sensors' temperature changes have been analyzed in Section 2.ii.a. The X axis of Fig. 28 is the temperature recorded by the sensors, while the Y axis of Fig. 28 is the relative temperature error (always positive) for those temperatures.

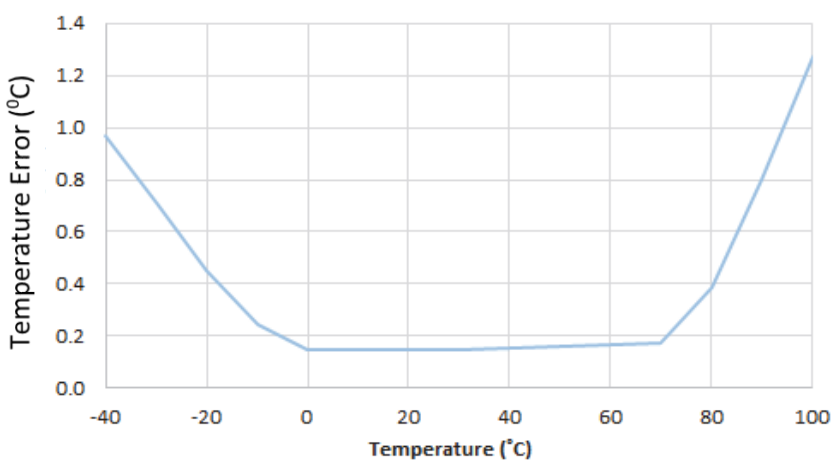


Fig. 28. Accuracy plot of sensor TMC6-HE over its temperature range (from Appendix B1)

2.i.b Flow Meters

Flow meters for pumps are briefly described herein, while Ref. 1 explains these in detail. There are four condensate pumps that are used in the power plant, and each pump is used for one week at a time. However, only pump #1 (i.e., the Worthington constant speed pump) and pump #2 (i.e., the Grundfos variable speed pumps) are analyzed in this thesis because these two pumps have data acquisition devices, and the other two pumps don't. To measure the flow rates of the water (0 - 528 gpm ± 0.25%) that these pumps discharge, flow meters were installed in the 4 inch discharge lines of the pumps. Schmidt [7] installed a Siemens electromagnetic flow meter (see Fig. 29) in each of the discharge lines of pumps #1 and #2. These flow meters have their own transmitters (Sitrans Mag 5000). The flow meters also have display screens which show the instantaneous flow rates of the water through the pipelines. The Sitrans Mag 5000 (see Appendix B2) transmits output signals (in the form of 4-20 mA) to an ONSET HOBO U12-006 Data Logger (see Appendix B3).

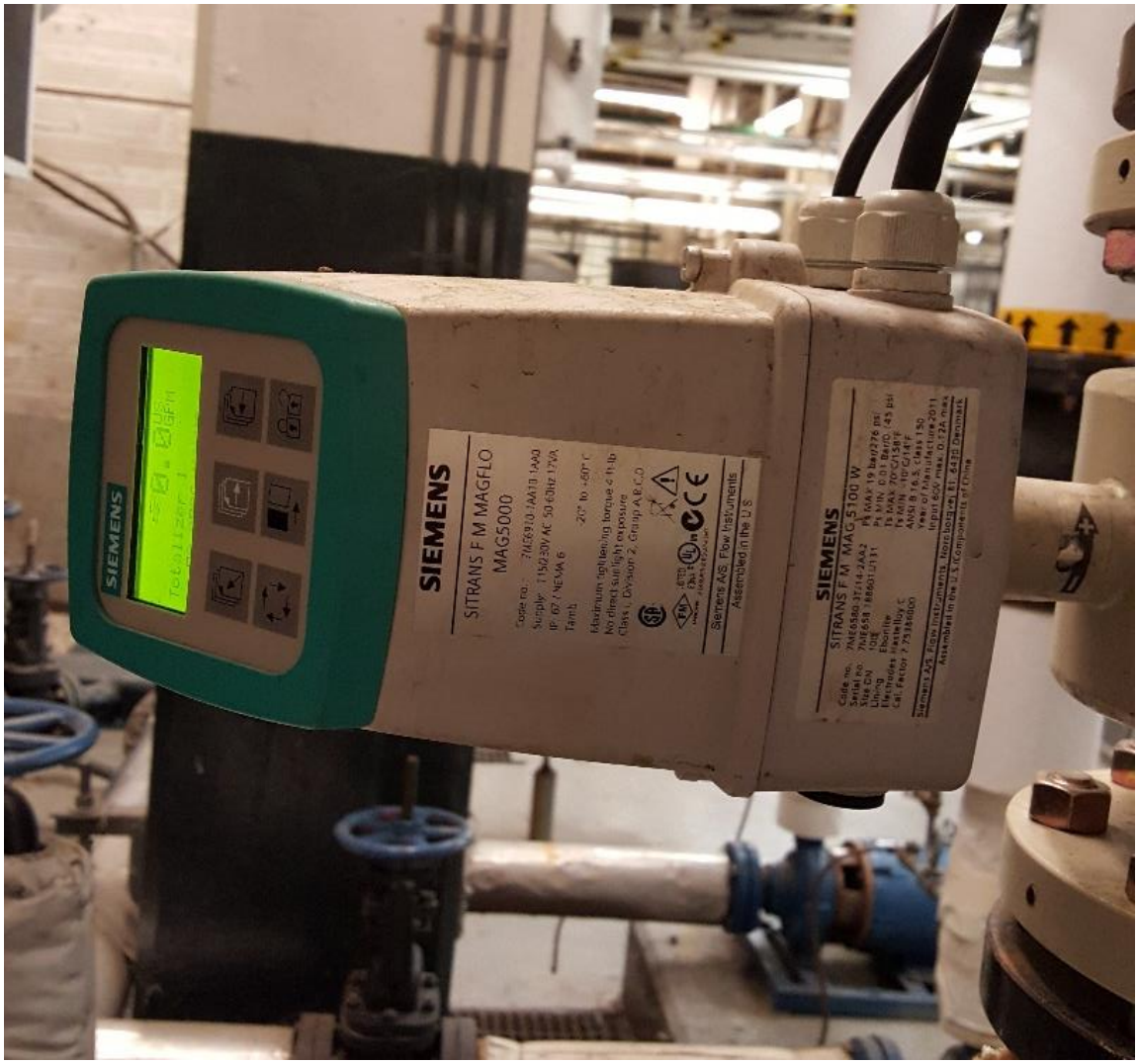


Fig. 29. Siemens MagFlow meter in the discharge line of condensate pump #2

A Siemens flow meter was also installed by Schmidt [7], with its associated transmitter, in the 1 inch water recirculation line of the pumps. As explained in Section 1.iii, this recirculation line is only used when there is a need to relieve excess flow in the discharge lines, so that water can be recirculated back to the inlets of the pumps (see Fig. 7a). Flow rate measurements through this pipeline have not been used for this thesis, because the valve allowing flow of condensate water through the recirculation line has always been closed.

There is also a flow meter installed in the 2 inch pipeline that goes to the vent condenser (see Fig. 30). This is a Cadillac Electromagnetic Meter (see Appendix B4); and it measures the flow rate of the water that goes into the vent condenser during normal power plant operation (see Fig. 7a).

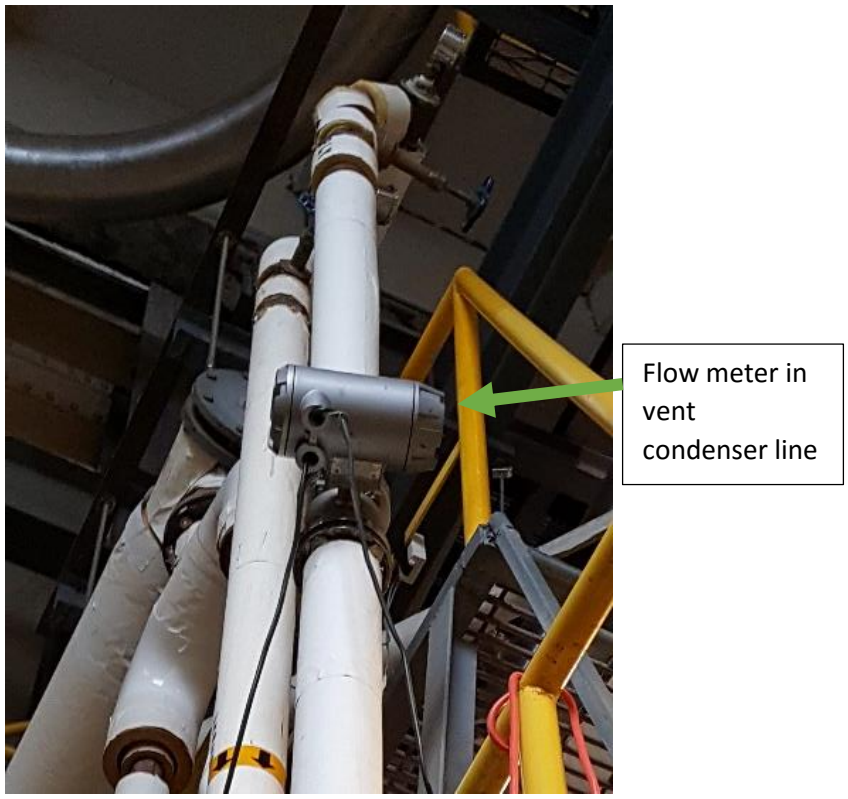


Fig. 30. Cadillac flow meter in the pipeline to the vent condenser on the 1st floor (just before the inlet of the vent condenser)

It was discovered in the summer of 2014 [1] that there were discrepancies in the measurements of these flow meters. At that time, the Cadillac flow meter read 93 gpm while the Siemens flow meter for the Worthington constant speed pump read 30.5 gpm. This was clearly not possible, because the Siemens flow meter (for the Worthington constant speed pump) measured the total condensate flow, while the Cadillac flow meter showed only the part of the condensate flow that went up to the vent condenser. In other words, the Siemens flow meter should have had readings higher than those of the Cadillac flow meter. For this reason, both of the Siemens flow meters (for the Worthington and the Grundfos pumps) were calibrated in November of 2014 [1]. After calibration, both of the Siemens flow meters (but not the Cadillac flow meter) were reinstalled and appeared to be showing accurate readings.

2.i.c Flow of Makeup Water

Flow meters are briefly described in this section, and Ref. 7 provides more details on these flow meters. The makeup water flows from the water softening system to the storage tanks, passing through the heat exchanger on the way (see Fig. 17). Makeup water flow rate could not be measured as there was no flow meter installed in the makeup water pipeline. Flow meter installation was proposed, but the power plant staff did not allow this installation because this would require closing down the power plant for a few hours. However, makeup water volume data is recorded by an existing meter (see Fig. 31) near the water softening units. This meter does not have a data acquisition system. The least count of the device is 10 gallons with an error of ± 5 gallons. The power plant staff record the total volume of water used every hour of the day (see Fig. 32).

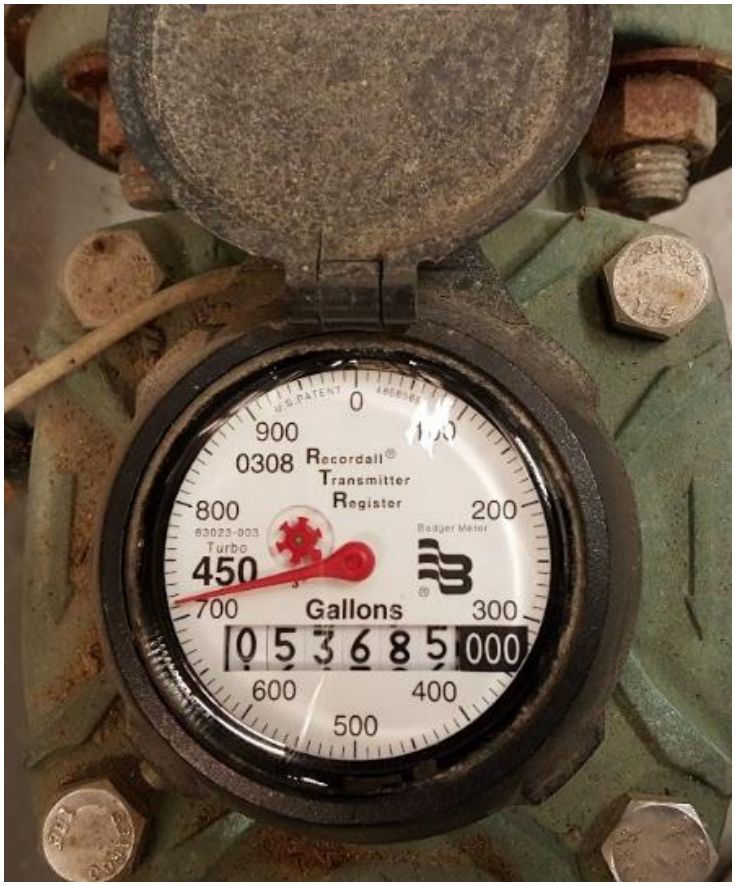


Fig. 31. Existing meter for makeup water

The volume for each hour was calculated by taking the difference between the readings of consecutive hours. Then, this hourly volume was divided by 60 to get the average volume used for each minute. However, this volume rate of usage is not constant, and hence further calculations were performed to determine the average flow rate in gpm. This is described in Section 3.iii. The results from these

approximate calculations of makeup water flow rate were used to estimate the energy savings of the heat exchanger in the basement of the power plant (see Appendix F).

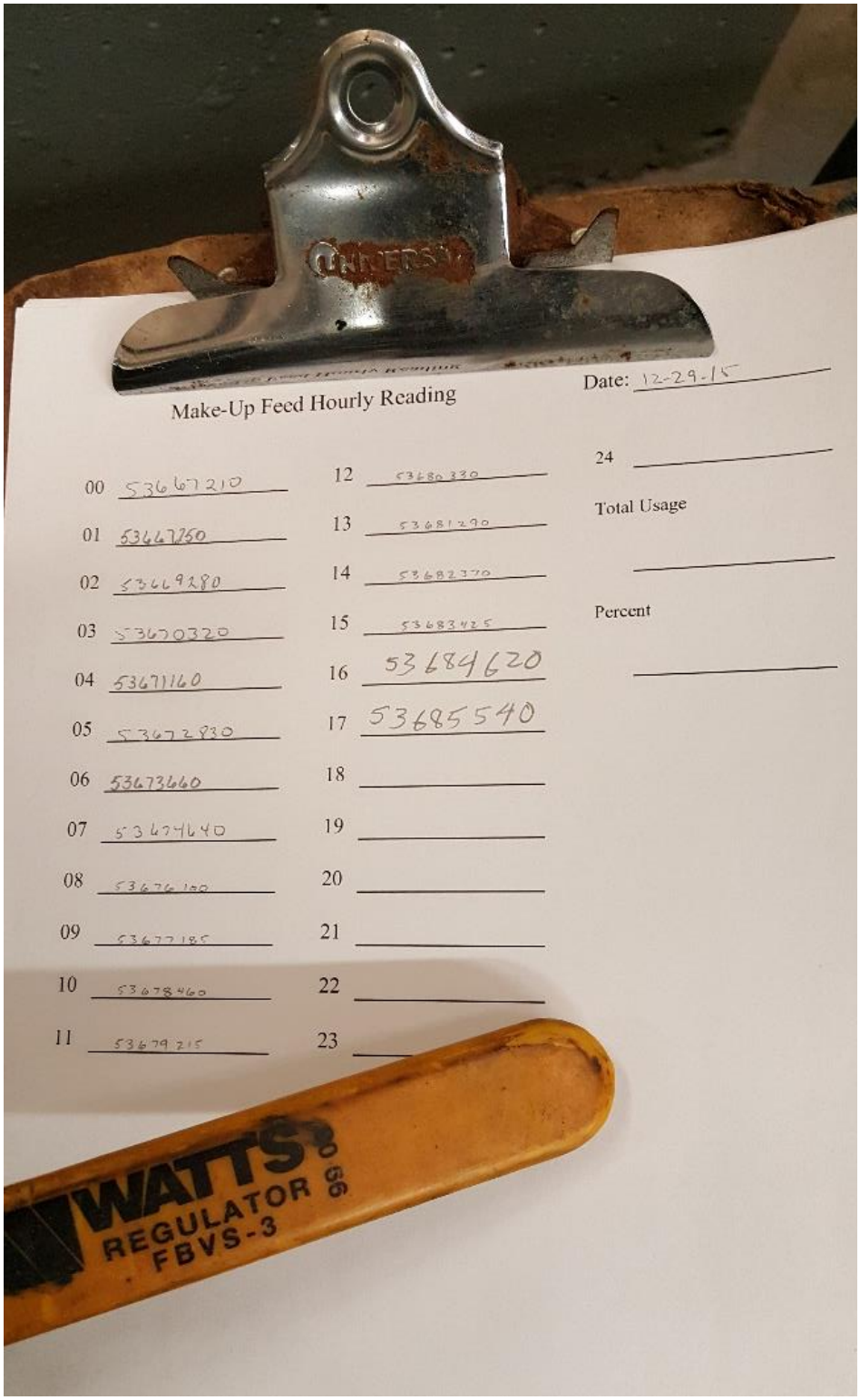


Fig. 32. Recorded hourly makeup water volume by power plant staff for December 29, 2015

2.i.d Pressure Transducers

Pressure transducers are briefly described in this section, and Ref. 7 provides more details on these transducers. To measure the discharge pressures from pump #1 (i.e., the Worthington constant speed pump) and pump #2 (i.e., the Grundfos variable speed pumps), Schmidt [7] installed pressure transducers in each of these pump's discharge lines. A Danfoss pressure transducer (see Appendix B5 for Danfoss MBS 3000) with a range of 0-145 psig ± 5% of full scale pressure reading, was installed in the pump #2 discharge line. This transducer (see Fig. 33) was coupled straight to the control panel of pump #2, and had two functions. First, the transducer served as the primary sensor for pump #2, making sure that it was running at the user specified pressure (usually 43 psig), when the pump was running in pressure control mode (set at the Grundfos control panel). Second, the transducer was also used for data acquisition when pump #2 was running in level control mode (set at the Grundfos control panel).

Fig. 33. Danfoss MBS 3000 pressure transducer in the discharge line (marked purple in Fig. 7a) of condensate pump #2 (marked blue in Fig. 7a)

Schmidt [7] also installed another pressure transducer on the suction side of pump #2. It is similar to the one that is in the discharge line of pump #2 (i.e., a Danfoss MBS 3000 transducer). However, its range of pressure measurement is much lower than that of the transducer for the discharge side (i.e., the range is 0-58 psig ± 5% of full scale pressure reading). This transducer was directly connected to the HOBO U12-006 Data Logger. The function of this transducer was to measure the suction pressure for pump #2, convert it into a 4-20 mA signal and transmit it to the data logger. Since the source of the water for both pumps #1 and #2 is the same (i.e., the condensate storage tanks), it was assumed that the suction pressure for pump #1 was the same as that of pump #2. Hence, a pressure transducer was not installed on the suction side of pump #1.

On the discharge side of pump #1, an Omega PX43E0-200GI pressure transducer (maximum pressure of 200 psig ± 5% of full scale pressure reading) was installed by Schmidt [7] (see Appendix B6). This transducer was also directly connected to the HOBO U12-006 Data Logger. The function of this transducer was to measure the discharge pressure of pump #1, convert it into a 4-20 mA signal and then transmit that signal to the data logger. However, this pressure transducer produced some negative pressure values during November and December of 2014. This was clearly not a possible situation because that would mean that there was a pressure less than atmospheric pressure in the discharge line of pump #1. Hence, the pressure transducer was replaced with an identical one in January of 2015 by Alabdullah [1]. It showed accurate pressure readings of 47-49 psig for pump #1.

Just before the control valve near the DA tank, there is another Danfoss pressure transducer MBS 3000 with a maximum pressure of 200 psig ± 5% of full scale pressure reading. This transducer measures the pressure in the pipeline just before the DA tank. The pressure drop from the condensate pumps to the inlet of the control valve can be computed using this transducer's readings and the readings from the transducers at the pumps' exits. There is also another pressure sensor, a Grundfos differential pressure sensor (0-2.5 bar ± 2% of full scale pressure reading) (see Appendix B7) that measures the pressure drop across the control valve. Both the Danfoss and the Grundfos pressure sensors are directly connected to a HOBO U12-006 Data Logger (see Appendix B2). The pressure transducers produce 4-20 mA output signals and transmit those signals to the data logger.

2.i.e Power Monitoring and Supply

To monitor pump #1's power, the pump has a Veris Power Monitoring H8044-0100-2 current transducer for power measurement (see Appendix B8). The purpose of this transducer is to continuously measure the current and voltage values of each of the three phases. From these values, the power consumption of the pump can be calculated. The transducer measures power at a maximum limit of 83 kW with an error of ± 1% of full scale power reading. This transducer is coupled directly to the same HOBO data logger as that for the pumps' pressure and flow.

The Grundfos pumps' control panel has a built-in power measuring device in the controller box. This device calculates the power consumption of the pump, by using the discharge flow rate and the discharge pressure that the pump produces, as input. Thus, an extra power monitoring device is not needed, assuming that the original Grundfos calibration stays accurate as the pumps age.

Most of the sensors (the current transducers, pressure transducers, etc.) require a power supply in order to run. They need an external voltage of 12-24 V DC. For this reason, two Mastech HY3003D DC power supply devices which supply 12-24 V DC are used (see Appendix B9): one is located near the condensate pumps (see Fig. 34), while the other is situated near the DA tank next to its control valve. The 4-20 mA output signals (see Fig. 35) are wired to the HOBO data loggers. These HOBO data loggers can store up to 65 megabytes of data and have four external channels, one channel for each 4-20 mA input. The Siemens magnetic flow meters have a different power supply, which they get directly from a 120 V AC power plant source.

Fig. 34. Mastech HY3003D DC power supply situated next to the condensate pumps (refer to Fig. 17 for approximate location in basement)

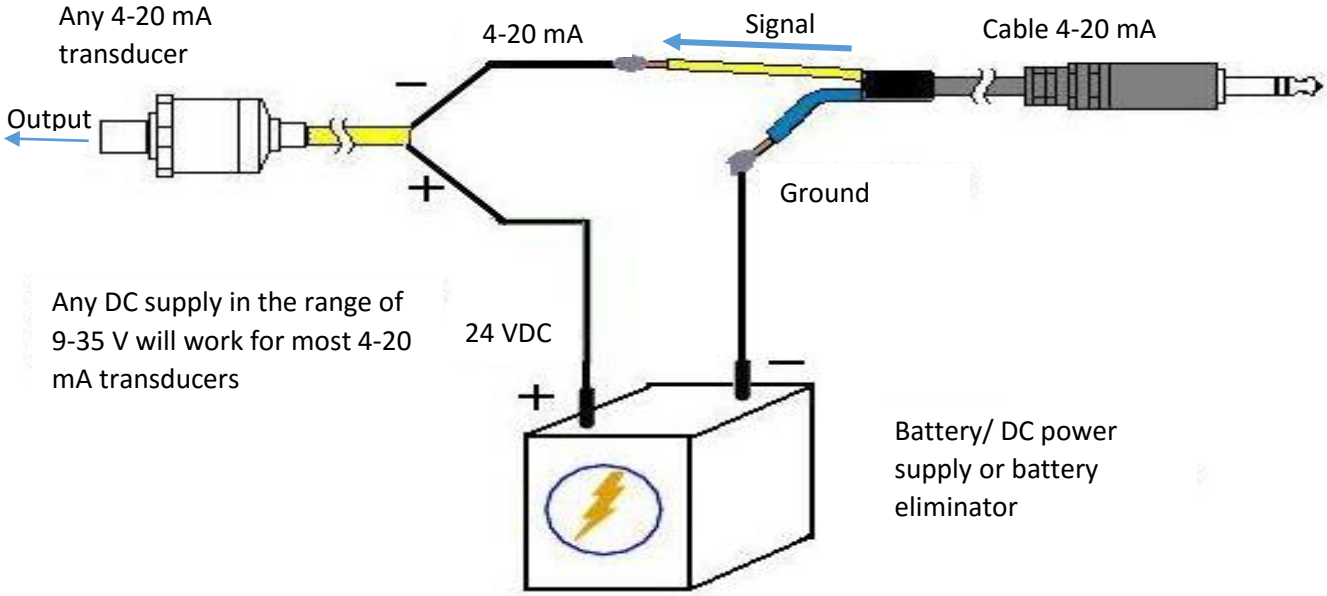


Fig. 35. Connections of 4-20 mA transducer to a DC power supply, with corresponding 4-20 mA output

2.i.f Level Control System

For running the Grundfos variable speed pumps in level control mode, a level sensor was required to determine the level of the water in the DA tank, because the Grundfos pumps did not have their own device to measure the water level in the DA tank. A standing pipe of the same height as the DA tank was installed by the power plant staff (see Fig. 36). A glass tube was connected to the standing pipe which

visually showed the elevation of the water inside the tank. For safety purposes, in case there was overflow of water or there was inadequate water in the DA tank, low and high water alarms were installed by the power plant staff (see Fig. 36).

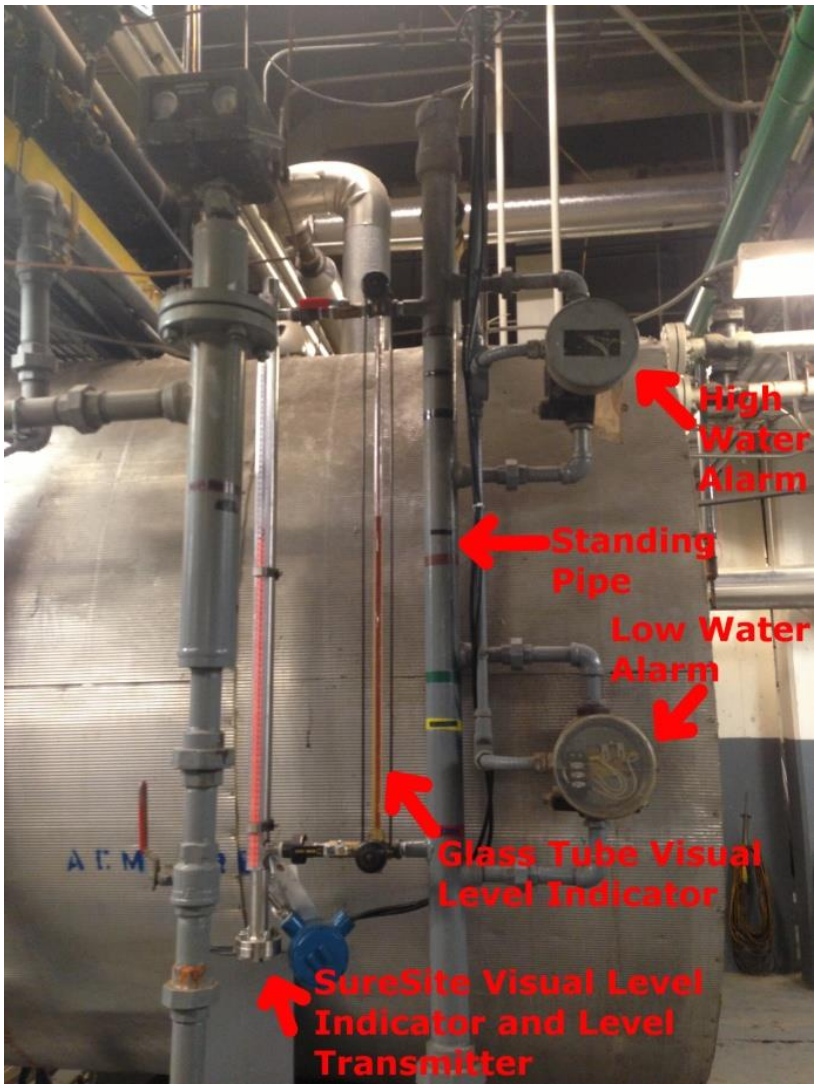


Fig. 36. Level control system attached to the DA tank in the basement

A SureSite visual indicator and level transmitter (see Appendix B10) was installed by Schmidt [7] next to the existing tube indicator as shown in Fig. 36. A DC Mastech Power supply (see Appendix B9) provides power for the SureSite sensor. The sensor is a magnet inside an aluminum casing that moves up and down according to the water level in the pipe. The water level in the pipe is the same as the water level in the DA tank. Pivoting "flags" are also present, that have two different colors: red and gray. The vertical movement of the magnet displays the height of water inside the casing, based on the color of the flag at that time: if the level of the water increases in the DA tank, the flags turn red; if the level of the water decreases in the DA tank, the flags go back to their original gray color.

The sensor is coupled to a calibrated transmitter which transmits a 4-20 mA output to the Grundfos pumps. The 4-20 mA output of the level transmitter is transformed into a 0-100% signal by the Grundfos pumps' control panel. For level control mode, 52% (or a level of water just higher than half of maximum height of the DA tank) was determined to be the ideal set point [7].

2.ii Data Acquisition Systems and Logging Procedure

This section covers the data acquisition systems and the data recording processes of those data acquisition systems. Temperature was recorded by two HOBO UX 120-006M data loggers (see Appendix B11), and this data was recorded on a per-minute basis. The discharge pressure, flow rate and power consumption of the Worthington pump was also recorded on a per-minute basis by a HOBO U12-006 data logger. For the Grundfos pumps, the only data that was recorded by the HOBO U12-006 data logger was the discharge flow rate. The power consumption and the discharge pressure data of the Grundfos pumps were recorded continuously by the Grundfos control panel.

2.ii.a Temperature

As explained in Section 2.i.a, the HOBO TMC6-HE temperature sensors (see Appendix B1) were installed on the outsides of the pipelines which served as the inlets and outlets of both of the heat exchangers (i.e., the vent condenser and the basement heat exchanger) in the power plant. For the temperature sensors, the recently obtained HOBO UX120-006M data loggers were used to log temperature data (see Appendix B11). After the temperature sensors were purchased, they were calibrated to ensure that they were accurate. The calibration procedure is explained in detail in Appendix C. After calibration, the percentage difference among all of the sensors was found to be a maximum of ±0.02%. Thus, the temperature readings from the sensors were considered to be accurate and have been used in this thesis for energy gain calculations.

2.ii.b Pumps

There were two HOBO U12-006 Data loggers (see Fig. 37 and Appendix B11) used in the plant. One was located near the condensate pumps, and the other was located near the control valve next to the basement DA tank. To read all of the recorded data, the HOBO software was installed on a Gateway laptop (see Appendix B12). The data logger was connected to the laptop via a USB cable, and all of the data could be downloaded to the laptop via the cable every week. All of the sensors had different settings, which could be changed by the user via the HOBO software. Since, the HOBO data logger received data in the form of a 4-20 mA signal, the signal had to be converted to appropriate units based on the data collected (e.g., psig for pressure and gpm for flow rate). The HOBO data logger provided a way to convert and scale the data accordingly (see Appendix D). After this scaling of values was set, a linear relationship was created by the HOBO software between the raw data input and its corresponding scaled output. Plots of the recorded data then could be seen according to this scaling in the main HOBO data window.

Data was gathered from June of 2015 to November of 2015. The data was recorded for all weather conditions - during the lowest demand (summer, i.e., June – August of 2015); moderate demand (fall, i.e., September-October of 2015) and highest demand (winter, i.e., November of 2015).

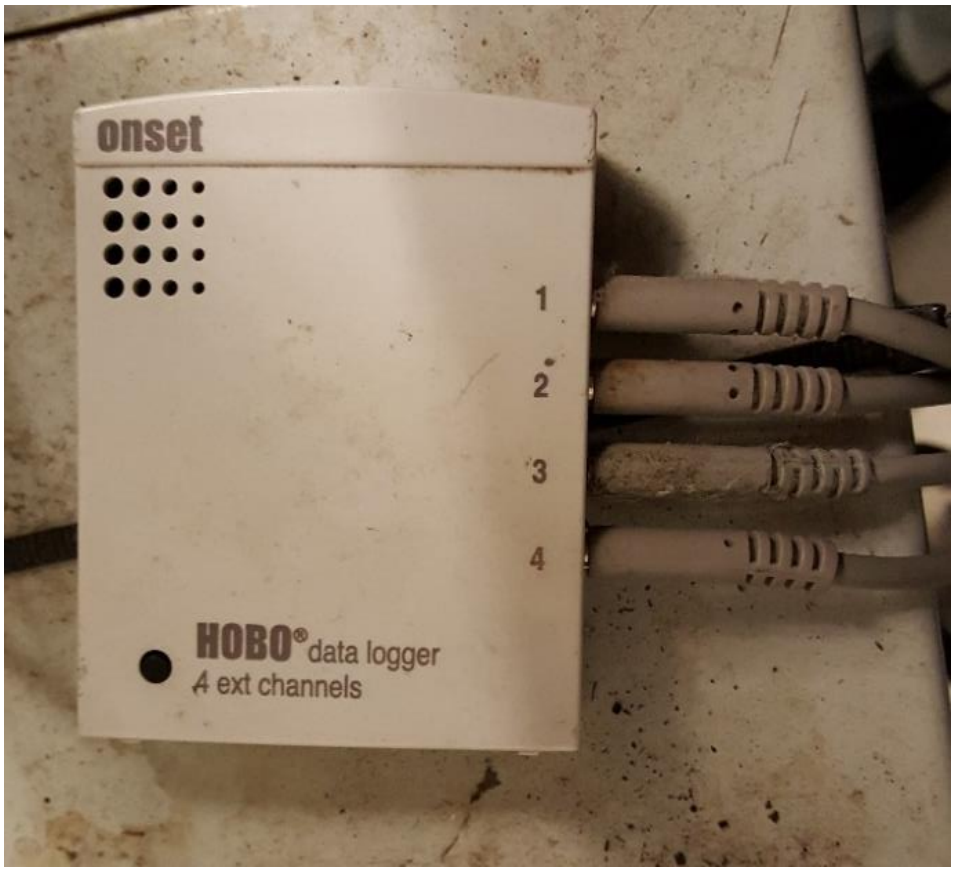


Fig. 37. HOBO U12-006 data logger next to the condensate pumps (see Fig. 17 for approximate location)

2.ii.b.i Worthington Constant Speed Pump

The following types of data (versus time) were collected for the Worthington constant speed pump (pump #1):

- Pump power consumption (kW)
- Pump flow rate (gpm)
- Pump discharge pressure (psig)
- Pressure before the control valve (psig)
- Pressure drop across the control valve (psig)
- Vent condenser water flow rate (gpm)

For this case, the HOBO data logger was used to record all of the previously listed data at one minute time intervals. Data was downloaded on a weekly basis to the Gateway laptop, as the power plant staff used the Worthington pump for one week every month from June to November of 2015.

2.ii.b.ii Grundfos Variable Speed Pumps

In this case, the following sets of data (versus time) were collected via the HOBO data logger:

- Pump flow rate (gpm)
- Pressure before the control valve (psig)
- Pressure drop across the control valve (psig)
- Vent condenser flow rate (gpm)

The PC-Tools E-Products software, supplied by the Grundfos Company, provided the following data (versus time):

- Pump flow rate (gpm)
- Pump discharge pressure (psig)
- Pump power consumption (kW)

The PC-Tools E-Products software was installed on the Gateway Netbook laptop (see Appendix B12). To record the three sets of data, the laptop was left running continuously in the KU power plant; and the software was restarted by the user every day so that the logged data would not be lost. Data was downloaded to the laptop every day of the week, as the power plant staff used the Grundfos pumps for one week of each month from June to November of 2015.

The Grundfos variable speed pumps were run in two different modes: Pressure Control and Level Control. Both of these cases have been investigated in this thesis; and the results are discussed in the next two sections.

2.ii.b.ii.a Pressure Control Mode

In this case, the Grundfos pumps were run at a constant discharge pressure throughout the entire week. The power plant staff set the discharge pressure at 43 psig. The control panel of the Grundfos pumps was used to set the discharge pressure. The condensate water that was discharged from the pumps went to the DA tank in the basement and to the vent condenser located on the first floor. The behavior of the Grundfos pumps, when run in pressure control mode, was almost identical to that of the Worthington pump. The data and results from the pressure control mode are covered in detail in Chapter 4.

2.ii.b.ii.b Level Control Mode

In this case, the Grundfos pumps supplied water to the DA tank only. The minimum pressure head required to push the excess condensate water to the vent condenser on the first floor was approximately 17.34 psig more than that required to reach the DA tank. The DA tank required water at a maximum pressure of approximately 10 psig. Because of the pumps' low discharge head (which varied between 10-25 psig), the pumps could not provide the extra 17.34 psig to push the excess water to the vent condenser. So, in this case, there was no flow of water to the vent condenser. Thus, in that situation, the Grundfos

pumps' only focus was to maintain the water in the DA tank at a desired level of 52% of the full DA tank capacity. The control valve before the DA tank was opened completely, allowing condensate water to freely flow into the DA tank. So, whenever the water level in the DA tank reached the desired set point, the Grundfos pumps reduced speed so as to save energy and also to make sure that water level did not rise much above 52% of the DA's full tank level.

For level control mode operation of the Grundfos pumps, the pumps' minimum performance point had to be set. The minimum performance point here is defined as: "the lowest speed at which only one of the Grundfos pumps can run (instead of both Grundfos pumps running all of the time) in order to conserve energy". Setting a minimum performance point for the Grundfos pumps required a lot of trial and error testing. 42% of full pump speed was determined as the ideal minimum performance point for the Grundfos pumps. This meant that, once the water in the DA tank reached its desired set point (i.e., 52% of tank capacity), the Grundfos pumps reduced their speed to 42% of full speed. Also, this meant that only one pump would run instead of both pumps at that speed. Then, if the DA tank required more water (i.e., when steam demand was higher), the Grundfos pumps started increasing speed. The pumps were programmed in such a way, that once 70% of full speed was reached, the second pump started running, and both pumps ran simultaneously at that point.

The Grundfos pumps allow the user to select the time interval setting (Ti) for the pumps' sensitivity of response to the DA tank's water level. Ti is the response time of the Grundfos pumps to the corresponding level of water in the DA tank. The range of Ti is from 0 to 10 seconds. The default Ti is 0.5 second. However, it was found that this response time was too fast, and thus a lot of power was wasted because there were sudden changes in discharge pressure and flow rate. By changing Ti to 2 seconds, there were still fluctuations in the data, but the amplitude and frequency were much smaller. This is explained in detail in Section 4.ii.b.

Running in Level Control mode for the Grundfos pumps was a challenge. Familiarity with the pumping system and the pressure drop in the lines was essential in order to carry out these tests. After changing the minimum pump performance (to 42% of full speed) and the time interval Ti (to 2 seconds), the pumps were able to provide condensate water to the DA tank without jeopardizing the DA tank and pipeline structure (i.e., without water "hammering"). Water hammering had occurred previously when the minimum pump performance was set at the default which was 60% of full speed and Ti was set at 0.5 second.

2.iii Averaging Pressure, Power and Flow Rate Data for the Grundfos Pumps

Since only the HOBO data logger was used for the Worthington pump, all of the recorded data values (i.e., discharge pressure, flow rate and power consumption) were averaged arithmetically because data was taken at equal time intervals (Δt) (i.e., one minute each). So, for this case, either arithmetic or time-weighted averaging gave the same result. When analyzing the Grundfos pumps, arithmetic averaging was used only for the data recorded from the Siemens flow meter because the flow rate was recorded by the HOBO data logger for equal Δt values of one minute each.

The PC-Tools E-Product software recorded data (discharge pressure, power consumption and also flow rate) at variable time intervals (Δt). Whenever there was a change in the data values, data was recorded. Thus, the time intervals (Δt) were continuously changing, i.e., the time intervals could be one second, five seconds, one minute or even one hour. Because of the non-uniform nature of the time intervals, arithmetic averaging was not an accurate method for analyzing the data. Thus, the time-weighted average [15] shown in Eq. (1) was used to calculate the average value for each recorded data segment from the PC-Tools software.

$$Q_T = \frac{1}{t_j - t_o}\int_{t_o}^{t_j} Q(t)dt \approx \frac{1}{2}\{(t_1 - t_o)\,[Q(t_o) + Q(t_1)] + (t_2 - t_1)[Q(t_2) + Q(t_1)] + \cdots + (t_j - t_{j-1})[Q(t_{j-1}) - Q(t_j)]\} \quad (1)$$

An example for flow rate data is shown in Fig. 38. The area under the curve was computed, then divided by the total time period (t_j-t_o) of the recorded data in order to calculate the average value (in this case, the flow rate). The terms used in Eq. (1) are shown in Fig. 38.

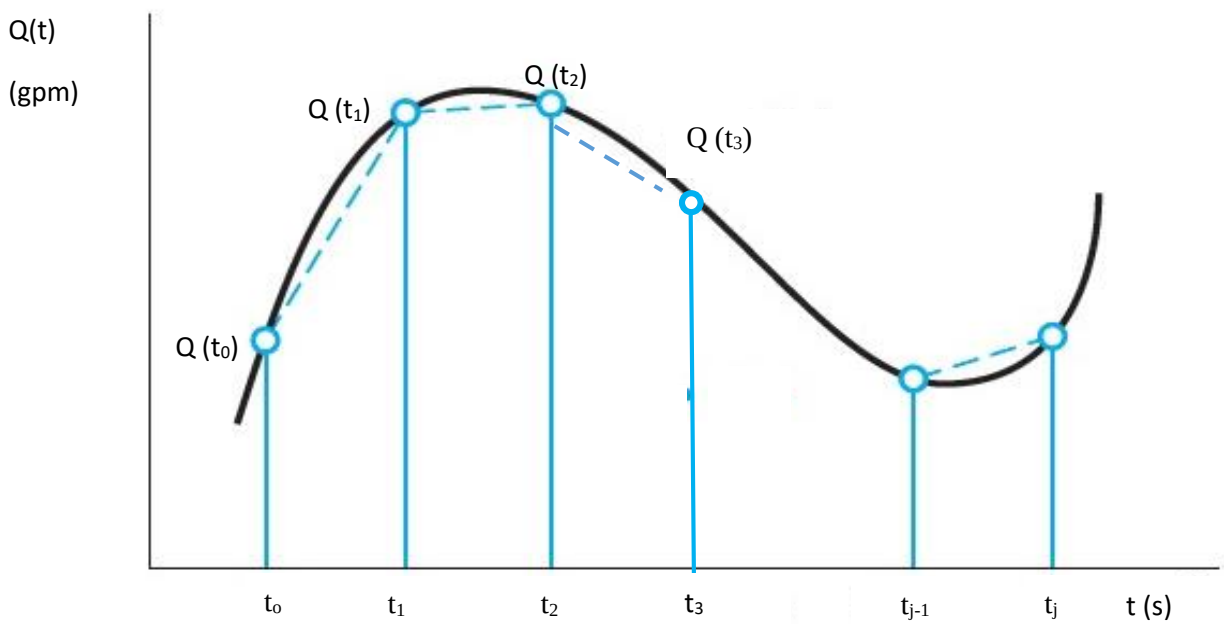


Fig. 38. Example plot of flow rate data with varying time intervals

The results of this approach are discussed and explained in Section 4.ii.

Chapter 3: Heat Exchanger Calculations

This chapter deals with the energy saved by using both of the heat exchangers in the KU power plant: the vent condenser on the first floor (for heating the excess condensate water) and the heat exchanger in the basement (for heating the softened makeup water). Approximate calculations were used to estimate the amount of energy that was reclaimed by the vent condenser and by the basement heat exchanger. Calculations were then done to estimate the amount of fuel (i.e., natural gas) that was saved by heating the condensate water before it was sent to the boilers.

The KU power plant's working model and the T-S diagram for the plant are shown in Figs. 7b and 7c. During process 1-1', the boiler feed water enters the economizer at a pressure of 175 psig in a compressed liquid state. During process 1'-2, the feed water enters the boiler in a saturated state and is converted into saturated steam at a constant pressure of 175 psig. The temperature during process 1'-2 is constant at approximately 377 ^{0}F. The steam produced by the boiler is throttled to a pressure of 90 psig, as shown by process 2-3. The enthalpy of steam remains the same (due to throttling) and the steam is in a superheated condition during process 2-3. This 90 psig steam is then circulated throughout the campus buildings. In process 2-4, some of the 175 psig superheated steam is bled to the DA after throttling through a number of valves to a pressure of approximately 10 psig, with a decrease in steam temperature from 377 ^{0}F to 315 ^{0}F (at state 4). The steam flows into the DA in process 4-6 at this temperature. After interacting with water in the DA, the steam loses some energy and its temperature in the DA drops to 225 ^{0}F. The vent condenser receives this steam and non-condensable gases from the DA to heat the excess condensate water, after throttling, during process 5-7. At state 7, the temperature of the steam is approximately 220 ^{0}F. The steam flows through the vent condenser during process 7-8, after which it is vented to the atmosphere.

The storage tanks also receive the condensed water in the return line from the campus at a temperature of approximately 165 ^{0}F and at a pressure of approximately 5-7 psig. Processes 9-12 and 10-12 show the water flowing into, and mixing in, the storage tanks from the vent condenser and campus return lines, respectively. State 19 is the makeup water source. This water is at a temperature of 60 – 75 ^{0}F, before it flows through the heat exchanger in the basement, during process 19-11. The basement heat exchanger heats the makeup water to a temperature in the range of 90 – 145 ^{0}F during process 19-11. The heated makeup water then goes to, and mixes in, the storage tanks (process 11-12), where it also mixes with the water from the vent condenser (process 9-12) and the campus return lines (process 10-12). The storage tanks' exit water temperature is approximately 160 ^{0}F.

During process 12-13, the condensate water is pumped by the condensate pumps. Process 13-14-6 shows the condensate water flowing into and through the DA after throttling via a control valve, thus reducing the pressure during the process. Process 13-9 shows the excess condensate water flowing through the vent condenser, where it is heated by steam and non-condensable gases (i.e., which the vent condenser receives from the DA, process 7-8); and then this heated water is returned to the storage tanks. There is not much change in pressure, as compared to the power plant pressures, in process 13-9, because the gate valve in this process is completely open to allow maximum flow of water to the vent condenser. Process 6-15 shows the water from the DA being pumped by booster pumps and boiler feed water pumps. At state 15, the pressure of the water is approximately 350 psig. Process 15-1 shows throttling from 350 psig to 175 psig (via regulator valves), and this 175 psig water goes through the economizer during process 1-1'.

All throttling processes in the T-S diagram have been represented with dashed lines. The arrows indicate the direction of flow of the water/steam. The entire boiler blowdown is represented by processes 16-17-18 in the form of red dashed and solid lines. The blowdown is actually not part of the typical T-S diagram; but it is shown so that all of the processess occuring in the power plant can be diagrammed in the same figure. The blowdown occurs from the boiler and undergoes throttling, as shown by process 16-17 by a red dashed line. In process 17-18 (red solid line), this water flows through the basement heat exchanger where its temperature gets reduced so that it is below 140 ^{0}F, after which the blowdown water is drained to the sewers.

3.i Procedure for Calculating the Reclaimed Energy from Heat Exchangers

Cold makeup water flows through the basement heat exchanger (see process 19-11 in Figs. 7b and 7c) before it goes to the condensate storage tanks (see process 11-12 in Figs. 7b and 7c). As explained earlier, the basement heat exchanger receives both the makeup water (that comes from the water softening tanks) and the hot water from the flash tank (the water that is condensed from the steam) in separate pipelines. Due to the exchange of heat between these two streams, the temperature of the makeup water increases, before it flows to the condensate storage tanks. Similarly, in the vent condenser on the first floor, the excess condensate water gets heated by the steam coming from the DA tank (see process 7-8 in Figs. 7b and 7c). The condensate water and the steam flow in separate pipelines in the vent condenser. The excess condensate water is then returned to the condensate storage tanks in the basement (see process 13-9-12 in Figs. 7b and 7c).

Because both the basement heat exchanger and the vent condenser help in transferring the heat from the hot water/steam to the relatively colder makeup/condensate water, energy is recovered or reclaimed. This energy would have otherwise been lost. This reclaimed energy is calculated by

$$E_{vent/HEX} = m_{vent/HEX}\, c_p\, \Delta T_{rise} \tag{2}$$

According to Alabdullah [1], the numerical value of temperature rise of the condensate water as it flowed through the vent condenser was found from randomly observing the temperature gauges across the vent condenser over four months. The average temperature rise across the vent condenser that Alabdullah recorded was 18.4^{o}F [1], which was approximated as 19°F in his calculations. This approximation was made based on the inlet and outlet temperatures that were visually taken from the analog temperature gauges.

Since digital temperature sensors were installed (see Section 2.i.a) after Alabdullah finished his work, the temperature rise is more accurately presented here. This temperature data was logged every minute so that the trends of the temperature change could be seen spanning several days. Sections 3.ii and 3.iii provide details on this new temperature data.

The energy transferred to the water by the power plant comes from burning natural gas. So, reclaiming energy from both the basement heat exchanger and the vent condenser results in burning less natural gas for heating the water to convert it to steam. Natural gas is provided to the KU power plant by the Black Hills Energy Company. Thus, the power plant reduces its budget by needing less natural gas when it uses both of the heat exchangers.

One of the terms required to calculate overall plant efficiency is the boiler efficiency. In order to calculate boiler efficiency, the amount of natural gas that would be needed to provide energy for the conversion of feed water to steam in the boiler is required. Boiler efficiency is a measure of the amount of combustion energy that can be converted into steam energy in the boiler. Boiler efficiency is defined as [16]

$$\eta_{boiler} = \frac{(E_{steam} - E_{BFW})}{E_{fuel}}$$

or

$$\eta_{boiler} = \frac{\left(m_{steam}\, h_{sat.steam} - m_{BFW}\, h_{water(BFW)}\right)}{Vol_{fuel}\left(LHV_{fuel}\right)} \qquad (3a)$$

The boiler efficiency equation should include only the amount of boiler feed water that is converted to saturated steam. However, some of that feed water exits the boiler in the form of boiler blowdown. This blowdown water has to be taken into account in Eq. (3a). Since there is no flow meter installed in either the main steam line or the blowdown line, it is not possible to calculate the exact amount of feed water that is converted to steam in the boiler or the flowrate of boiler blowdown. Thus, it is assumed that that steam is produced directly by 98% of the boiler feed water while 2% of the boiler feed water is lost as blowdown [4].

The m_{steam} shown in Eq. (3a) refers to all generated steam from the boiler, which includes the amount of steam extracted from the main steam line and injected into the DA tank in order to preheat the boiler feed water in the DA tank. m_{BFW} refers to the boiler feed water that is converted into steam in the boiler. According to the assumption that 98% of m_{BFW} is considered to approximately equal the amount of generated steam from the boiler, i.e., $98\ \%\ m_{BFW} \approx m_{steam}$, Eq. (3a) can be rewritten as

$$\eta_{boiler} = \frac{\left(m_{steam}\, h_{sat.steam} - \frac{m_{steam}}{0.98}\ h_{water(BFW)}\right)}{Vol_{fuel}\left(LHV_{fuel}\right)} \qquad (3b)$$

All terms in Eq. (3b) are defined in the Nomenclature.

Since the fuel used in the steam power plant is natural gas, an average Lower Heating Value (LHV_{fuel}) of natural gas is employed which is equal to 1018.6 Btu/ft^3 for 7 inch water gauge gas pressure and a temperature of approximately 60 ^{0}F. This information was obtained from George Werth, KU's Campus Energy Engineer [17].

The natural gas fuel flow rate (Vol_{fuel}) in Eq. (3b) is provided by the steam power plant's operating log sheets [4] (see Appendix E). m_{steam} in Eq. (3b) is also given by those log sheets (see Appendix E). The enthalpies for both the saturated steam ($h_{sat.steam}$) and the boiler feed water ($h_{water(BFW)}$) are found from the steam tables [16], as explained in the following discussion.

The BFW temperature is taken from averaging the maximum and minimum temperatures of the boiler feed water flow. The BFW enters the economizer as a compressed liquid (State 1 in Figs. 7b and 7c). So, the enthalpy of the boiler feed water is found from the steam tables for compressed liquid conditions. From observation of the BFW temperature in the hourly log sheets [4], the maximum and the minimum BFW temperatures are found to be 230 ^{o}F and 220 ^{o}F. Therefore, the average boiler feed water

temperature is taken as 225 °F. At this boiler feed water temperature in compressed liquid conditions, the enthalpy is 193.3 Btu/lb_m (from the steam tables at 175 psig and 225 °F). For $h_{sat.steam}$, 175 psig saturated steam has an enthalpy of 1197.7 Btu/lb_m (from the steam tables, also at 377 °F).

Finally, using this information in Eq. (3b), the average daily and monthly boiler efficiency can be calculated (see Appendix E). Even though the daily boiler efficiency information is given in the log sheets provided by the steam power plant, this efficiency is not used in the calculations in this thesis because there is no explanation for how the log book boiler efficiency is calculated.

For calculating the overall plant efficiency, the energy reclaimed by both heat exchangers with respect to the combustion energy of the natural gas in the boilers, has to be taken into account. The plant efficiency can then be calculated by including both the boiler efficiency and the energy reclaimed by both heat exchangers with respect to the natural gas combustion energy in the boilers.

$$\eta_{plant} = \eta_{boiler} + \frac{E_{vent/HEX}}{Vol_{fuel}(LHV_{fuel})} \tag{4a}$$

Because the vent condenser and the basement heat exchanger heat the boiler feed water to a higher temperature before the water reaches the boiler, less natural gas is required in the boiler to convert the feed water into steam. In other words, natural gas is saved by using either the vent condenser or the basement heat exchanger. The volume of natural gas saved is calculated by

$$Vol_{fuel,saved} = \frac{E_{vent/HEX}}{\eta_{plant}\,(LHV_{fuel})} \tag{4b}$$

See Appendix F for the energy gain calculations of both heat exchangers.

3.ii Vent Condenser on the First Floor

For the vent condenser, the Cadillac flow meter (see Appendix B4) accurately logged the flow rate, which was used in the calculation of natural gas saved. The TMC6-HE temperature sensors (see Appendix B1) recorded the temperatures (with an error of ± 0.36 ^{0}F to ±0.72 ^{0}F) of the condensate water at the inlet and at the outlet of the vent condenser on a per-minute basis, for the months of October and November of 2015 (see Figs. 39 and 40). The temperature rise in the vent condenser (with an error of ±0.72 ^{0}F to ±1.44 ^{0}F) for the months of October and November is shown in Figs. 41 and 42.

It was found that approximately 353,800 ft^3 of natural gas was saved in October, while, in November, approximately 476,980 ft^3 of natural gas was saved (see Table F4) as compared to the case when the vent condenser was not used for both months.

Using the standard cost of natural gas (\$0.0053/ft^3) for 7 inch water gauge gas pressure and a temperature of approximately 60 ^{0}F (i.e., approximately ground temperature) [17], the vent condenser saved approximately \$1,875 in October and \$2,530 in November as compared to the case when the vent condenser was not used. See Appendix F for more details on these calculations.

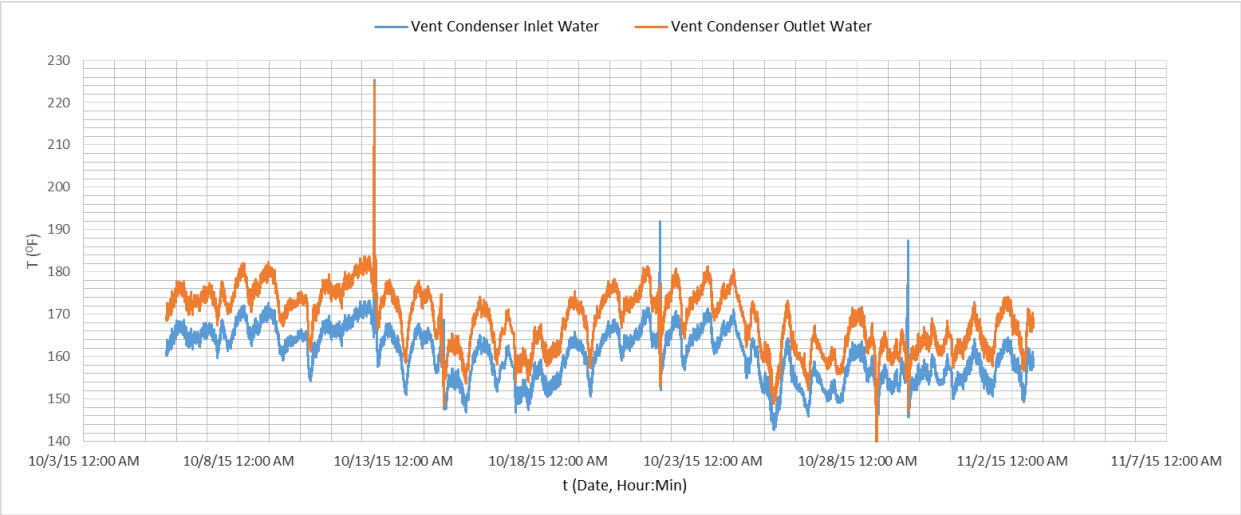


Fig. 39. Temperature of condensate water at the inlet and outlet of the vent condenser (for October of 2015)

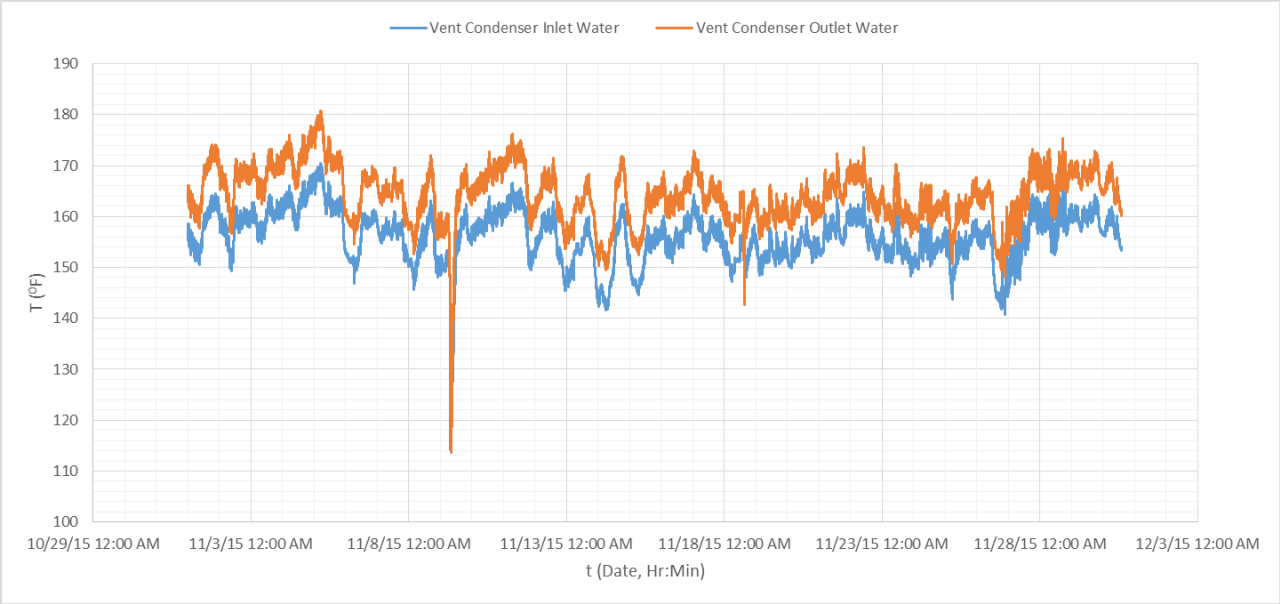


Fig. 40. Temperature of condensate water at the inlet and outlet of the vent condenser (for November of 2015)

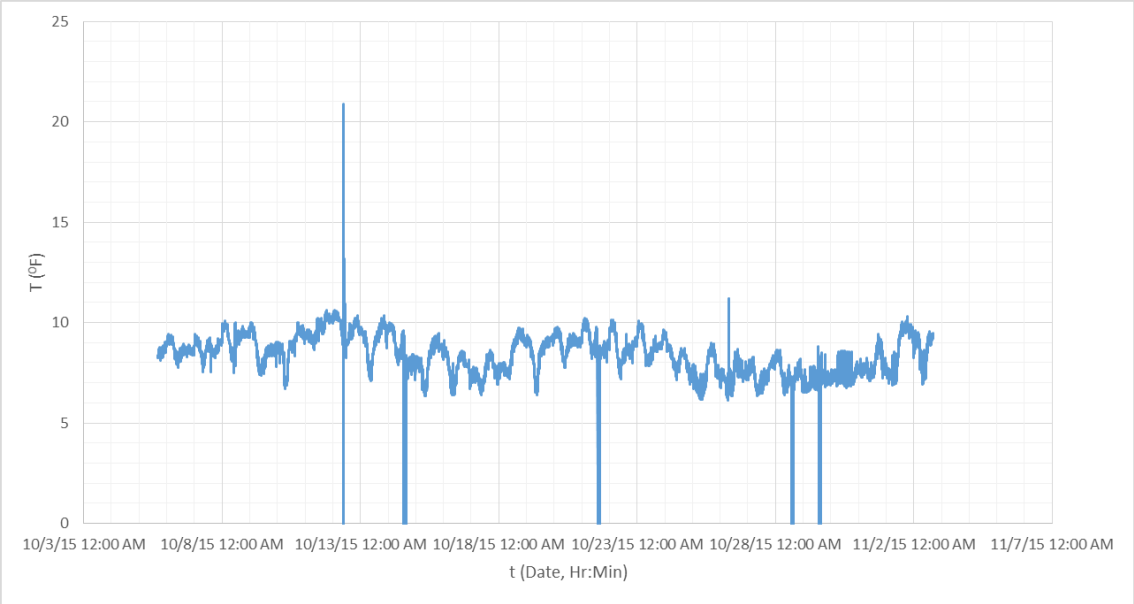


Fig. 41. Temperature rise of condensate water flowing through the vent condenser (for October of 2015)

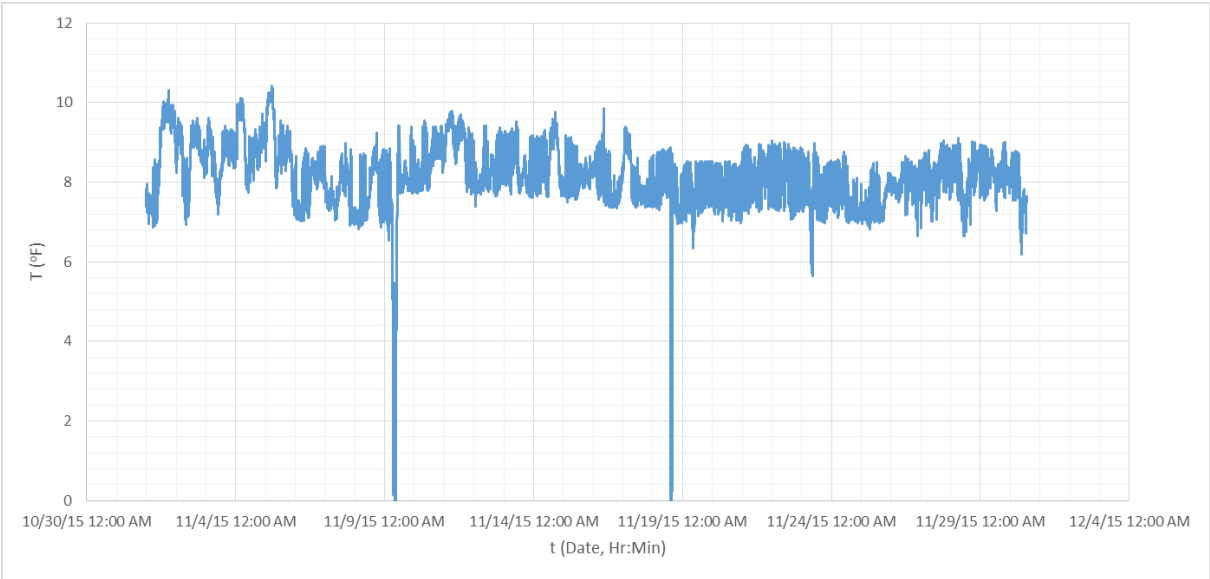


Fig. 42. Temperature rise of condensate water flowing through the vent condenser (for November of 2015)

Based on the amount of steam generated, the energy reclaimed by the vent condenser in every month was estimated (see Appendix F). The total energy reclaimed by the vent condenser for the year of 2015 was estimated to be 4,790,000,000 Btu (see Table F3) as compared to the case when the vent condenser was not used. Therefore, the estimated savings for natural gas in calendar year 2015 was found to be $27,680 as compared to the case when the vent condenser was not used. All of the calculations are shown in detail in Appendix F.

3.iii Heat Exchanger in the Basement

The HOBO temperature data logger was set to record the temperatures of the water in the two inlet pipelines and the two outlet pipelines. Temperatures were recorded on a per-minute basis for the months of October and November of 2015.

The temperature rise of the makeup water exiting the heat exchanger was very inconsistent (see Fig. 43). It fluctuated from as low as 5 ^{0}F to as high as 140 ^{0}F. This is due to the pipelines sometimes being closed off. At the makeup water outlet, there is a mechanical valve. The power plant staff open the valve by varying amounts, based on the level of water in the condensate tanks. The condensate tanks have a level indicator, which the power plant staff check before they open/close the mechanical valve. The desired level of water in the condensate storage tanks is about 52% of the tanks' capacities. There is a control system attached to the storage tanks with a solenoid valve for each tank. The solenoid valve has two positions only: open or closed. Based on the water level in the DA tank, the control system sends a signal to the solenoid valve which then opens or shuts automatically. In this way, the solenoid valve controls the flow of makeup water into each storage tank from the heat exchanger. In Fig. 43, the temperature of the water at the inlet and outlet of the heat exchanger in the basement from 3:57 pm to 5:02 pm on October 5, 2015 is shown. In Fig. 44, the temperature rise of the makeup water as it passed through the heat exchanger from 3:57 pm to 5:02 pm on October 5, 2015 is shown.

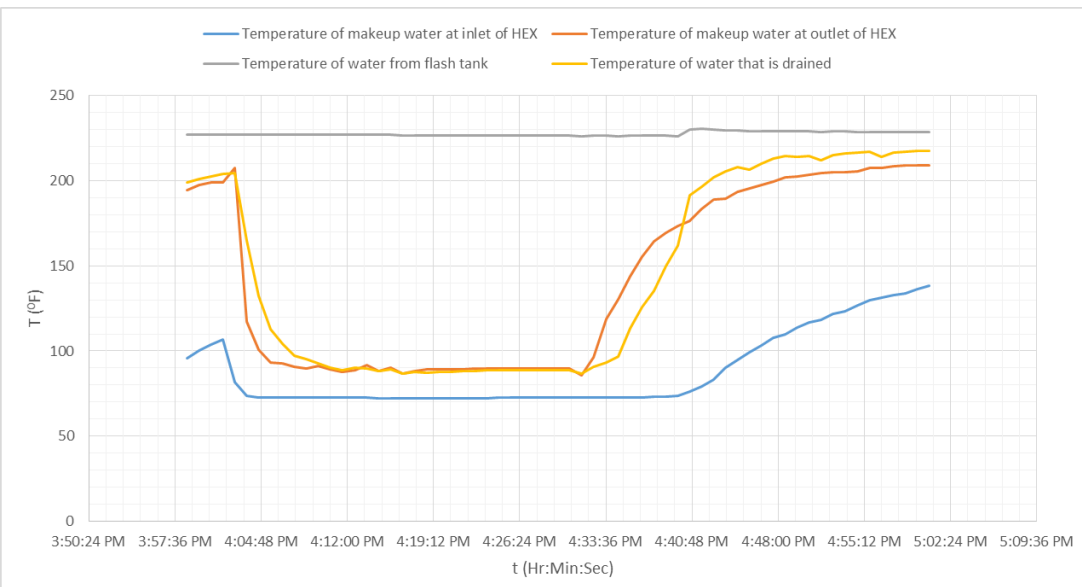


Fig. 43. Temperature of water at the inlets and outlets of the basement heat exchanger from 3:57 pm to 5:02 pm on October 5, 2015

When the water level in each condensate tank is more than 52%, the solenoid valve (on receiving signal from the control system) for that tank shuts off. For this case, the makeup water in the entire pipeline is stationary. Because this water is not moving, the temperature difference rapidly increases, and sometimes it reaches as high as 140^0 F. A test was conducted on October 5, 2015 between 6:20 pm and 7:30 pm (see Fig. 45) to see the change in temperature of the water based on either opening or closing of the mechanical valve. The temperature of the water was logged on a per-second basis.

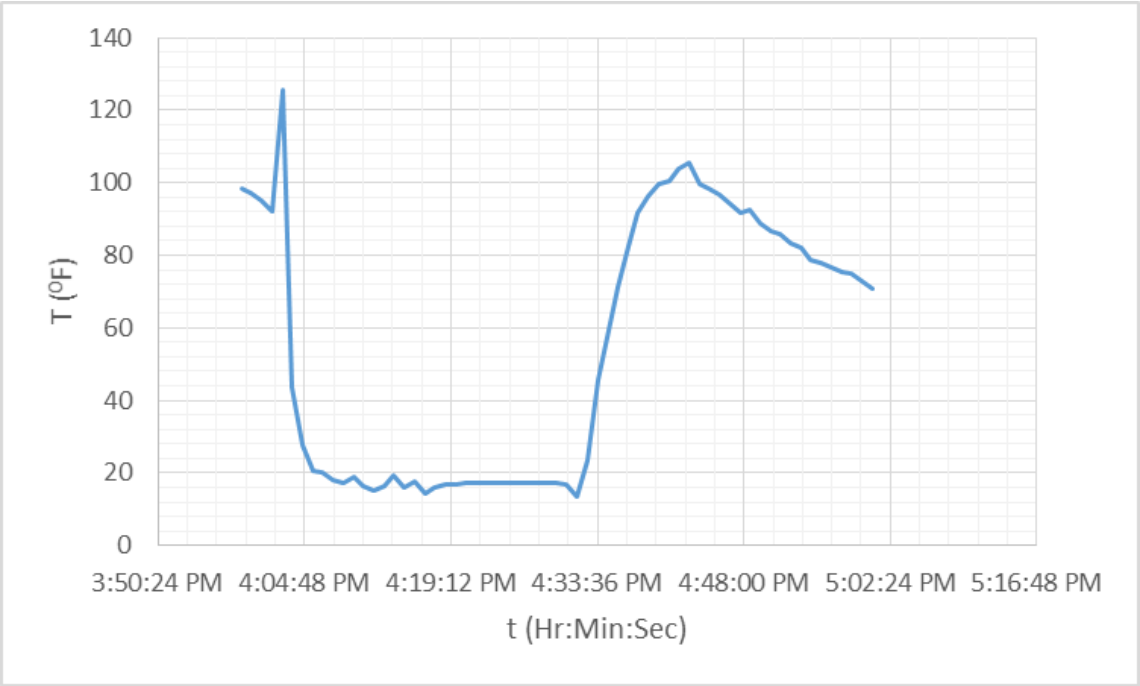


Fig. 44. Temperature rise of makeup water flowing through the basement heat exchanger from 3:57 pm to 5:02 pm on October 5, 2015

At the beginning of the test, the mechanical valve was initially open (see Fig. 45). The mechanical valve was then completely closed. Once the mechanical valve was closed, the temperature of the makeup water started to gradually rise. This was because there was no flow of makeup water through the heat exchanger. Also, the solenoid valves just before the condensate storage tanks were open during this time interval. After some time, the mechanical valve was opened 100%. At the beginning of this situation, there was a drastic drop in temperature of the makeup water. However, after some time, the temperature started gradually rising, even though the mechanical valve was 100% open. This was because the solenoid valves shut off completely, which meant that the water level in the condensate tanks was more than 52% of tank capacity. Thus, there was no flow of makeup water in the heat exchanger during this time interval. It was thus concluded that the makeup water flow was highly dependent on the solenoid valve, regardless of the mechanical valve being open/closed. Even though power plant staff tried to minimize these high temperature rises in the makeup water by opening/closing the mechanical valve, it was not a foolproof method because of the independent nature of the solenoid valve.

The temperature data was investigated for a month on a per-minute basis. It was seen that the temperature difference across the heat exchanger (for the makeup water) varied from 5 ^{0}F to 140 ^{0}F. Even though 35 ^{0}F was initially considered to be too high for the makeup water temperature rise, it was seen that the temperature rise in the makeup water remained constant at 35 ^{0}F for a 1-3 hours on October 7, October 9, October 15 and October 24 of 2015, when the solenoid valve was open. When the solenoid valve shut automatically, the magnitude of temperature rise immediately increased, going as high as 180 ^{0}F. As a result of these investigations, only temperature rises from 0 ^{0}F to 35 ^{0}F were considered for calculations of heat exchanger energy savings. All of the temperature rises above 35 ^{0}F were neglected, as it was assumed that water was not flowing in the pipe, but was just stationary for these situations.

Makeup water flow rate was also required for the calculation of energy recovered from the heat exchanger. However, the volume measuring device used by the power plant staff did not have a data acquisition system associated with it. Instead, hourly readings of the volume of makeup water were recorded by the power plant staff (see Section 2.i.c). So this data was employed.

Initially, the hourly readings were divided by 60 to give an average flow rate of makeup water in gpm. However, this was not accurate for measuring flow rate, because there were periods of time when there was no flow of makeup water, as explained earlier. So, an estimate of makeup water flow rate was made based on the temperature rise of the makeup water. This estimate assumed zero flow rate when the temperature rise was greater than 35 ^{0}F. The procedure for this process is detailed in the next few paragraphs and Tables 1 and 2.

The temperatures of the makeup water at the inlets and outlets of the basement heat exchanger were recorded by the HOBO data logger for every minute, and the temperature rise of the makeup water was subsequently calculated. Table 1 shows the temperature rise of the makeup water as it flowed through the heat exchanger between 12 – 1 am on October 6, 2015 (for 30 readings only, not 60).

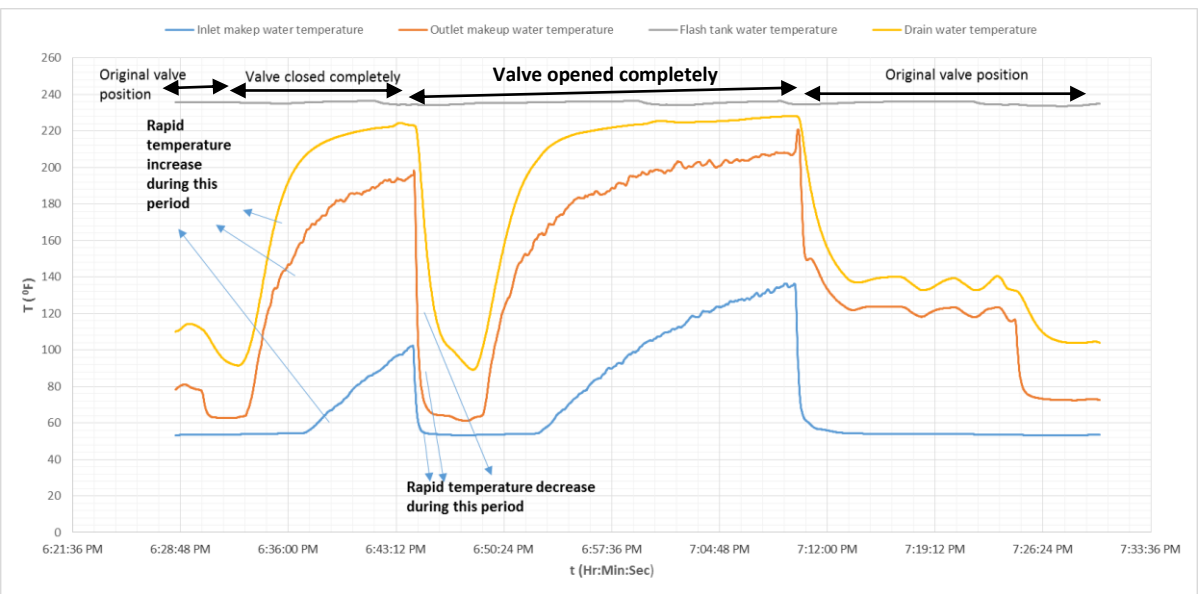


Fig. 45. Temperature of water at the inlets and outlets of basement HEX during test for opening/closing mechanical and solenoid valves on October 5, 2015

It was found that there were several time intervals (30 to be exact) within that hour that the temperature rise of the makeup water exceeded 35^0 F. It was assumed that the solenoid valve was closed during these time intervals and so, there was no flow of water through the heat exchanger (i.e., the water was stationary). Thus, the temperature rise recordings for these time intervals were not included in Table 1. This meant that there was flow of makeup water through the heat exchanger for the remaining 30 minutes of that hour, where the temperature rise varied from approximately 13^0 F to 33^0 F. The temperature differences for only those 30 time intervals were considered for further calculations of average flow rate.

Thus, the recorded hourly usage, 430 gallons for 12-1 am on October 6, 2015, was divided by 30 to get an initial/overall average flow rate of 14.333 gpm for those 30 minutes (which were not necessarily sequential in time).

Although the overall average flow rate was computed as 14.333 gpm, this was not the actual per-minute makeup water flow rate (for those 30 minutes). The temperature rise of the makeup water for every minute was not the same. It was therefore assumed that the flow rate would also not be the same for every minute. To estimate the makeup water flow rate for every minute, the temperature rises for the makeup water for each of the 30 minutes of non-zero flow were rearranged. It was assumed that the higher the rise in makeup water temperature, the lower the makeup water flow rate (i.e., an inverse relation). The temperature rise for every one minute time interval was inversely ratioed to the highest temperature rise recorded (in this case, 33.40 ^{0}F; row 30 of Table 1) (refer to 2nd column in Table 2). The 14.333 gpm average flow rate was then multiplied by this inverse ratio (refer to 3rd column in Table 2). The new flow rate value for every minute (or reading) was assumed to correspond to the relative temperature rise of the makeup water for that minute. Table 2 shows the decrease in the temperature rise values (refer to 1st column in Table 2) corresponding to increase in the flow rate values of makeup water (refer to 3rd column in Table 2) for the 30 time intervals.

The total hourly usage of the makeup water was 430 gallons. However, the sum of the estimated makeup water flowrates from column 3 of Table 2 did not equal 430 gallons for that hour because the procedure to get column 3 of Table 2 did not maintain consistent total overall flow. Instead, the sum of the estimated makeup water flow rates equaled 890.85 gallons for that hour. So, for the per-minute flow rates to sum up to 430 gallons for that hour, the estimated flow rates were reduced by a factor of 0.4826 (i.e., 430/890.85 = 0.4826). The new flow rates are shown in column 4 of Table 2.

Appendix F shows the calculation of the energy savings using the data from Table 2. Because of the extremely lengthy nature of computing the estimated makeup water flow rates (since the flow rates had to be computed for every hour for an entire month), these flow rates were computed on an hourly basis for two days in October (October 8 and October 21) and two days in November (November 8 and November 21) of 2015. The monthly savings were then projected based on those four days of data.

For the basement heat exchanger, Appendix F shows an estimated savings of 97,960 ft^3 of natural gas for October, while for November, approximately 132,060 ft^3 of natural gas was saved (see Table F4 in Appendix F). Based on the cost of natural gas being \$0.0053 per ft^3 [17], approximately \$520 was saved in October and approximately \$700 was saved in November as compared to the case when the basement heat exchanger was absent for both months.

The energy reclaimed by the basement heat exchanger for each month of 2015 was estimated based on the steam generated by the boilers for that month (see Appendix F for detailed explanation). The total energy reclaimed by the heat exchanger in the basement in 2015 was estimated to be 1,325,000,000 Btu. Thus, the basement heat exchanger's total savings for natural gas was estimated to be \$7,660.

Note that this approximate procedure was used to obtain an estimate of the savings as compared to the case when the basement heat exchanger was not used; but assuming a different maximum temperature rise than 35 ^{0}F would yield different results.

Detailed calculations of the basement heat exchanger's natural gas savings are shown in Appendix F.

Table 1. Temperature rise and corresponding flow rate of makeup water for 30 readings for which $\Delta T_{rise} \leq 35\ ^0F$

Time Intervals (min)	Date & Time	Temperature Rise (^0F)
1	10/06/15 12:04:36 AM	25.33
2	10/06/15 12:05:36 AM	20.62
3	10/06/15 12:06:36 AM	17.17
4	10/06/15 12:07:36 AM	13.56
5	10/06/15 12:08:36 AM	17.60
6	10/06/15 12:09:36 AM	14.16
7	10/06/15 12:10:36 AM	18.75
8	10/06/15 12:30:36 AM	29.30
9	10/06/15 12:31:36 AM	19.42
10	10/06/15 12:32:36 AM	18.48
11	10/06/15 12:33:36 AM	14.63
12	10/06/15 12:34:36 AM	19.64
13	10/06/15 12:35:36 AM	13.62
14	10/06/15 12:36:36 AM	18.73
15	10/06/15 12:37:36 AM	14.02
16	10/06/15 12:38:36 AM	15.18
17	10/06/15 12:39:36 AM	17.10
18	10/06/15 12:40:36 AM	13.04
19	10/06/15 12:41:36 AM	14.84
20	10/06/15 12:42:36 AM	17.04
21	10/06/15 12:43:36 AM	13.32
22	10/06/15 12:44:36 AM	14.13
23	10/06/15 12:45:36 AM	16.60
24	10/06/15 12:46:36 AM	14.81
25	10/06/15 12:47:36 AM	13.08
26	10/06/15 12:48:36 AM	15.97
27	10/06/15 12:49:36 AM	13.36
28	10/06/15 12:50:36 AM	15.44
29	10/06/15 12:51:36 AM	19.29
30	10/06/15 12:52:36 AM	33.40

Table 2. Ratios of temperature rises and corresponding flow rates of makeup water

Temperature rise (^{0}F)	Ratio of temperature rise	Estimated makeup water flow rate based on temperature rise (gpm)	Ratioed makeup water flow rates using conversion factor (gpm)
In descending order (X)	Y = (33.40/X)	(14.333 gpm) (Y)	Conversion factor = (430/890.95) = (0.4826)
33.40	1	14.33	7.07
29.30	1.13	16.71	8.06
25.33	1.31	19.33	9.33
20.62	1.61	23.75	11.46
19.64	1.70	24.93	12.03
19.42	1.71	25.22	12.17
19.29	1.73	25.39	12.2
18.75	1.78	26.11	12.60
18.73	1.78	26.15	12.62
18.48	1.80	26.50	12.79
17.60	1.89	27.83	13.43
17.17	1.94	28.52	13.76
17.10	1.95	28.64	13.82
17.04	1.95	28.74	13.87
16.60	2.01	29.50	14.24
15.97	2.09	30.67	14.80
15.44	2.16	31.70	15.30
15.18	2.19	32.25	15.56
14.84	2.25	33.00	15.93
14.81	2.25	33.06	15.96
14.63	2.28	33.46	16.15
14.16	2.35	34.58	16.69
14.13	2.36	34.64	16.72
14.02	2.38	34.93	16.86
13.62	2.45	35.94	17.35
13.56	2.46	36.11	17.43
13.36	2.49	36.64	17.68
13.32	2.50	36.75	17.74
13.08	2.55	37.43	18.06
13.04	2.55	37.54	18.12
Sum of values →		890.68	430

3.iv Calculation of Estimated Heat Exchanger Area

The two heat exchangers in the KU power plant are shell-and-tube type. These heat exchangers are designed so that they have an outer shell enclosing a series of tubes. One fluid at a certain temperature flows through the tubes, and a second fluid, which is at a different temperature than the first fluid, flows over and around the tubes (i.e., this fluid flows through the shell). This results in heat transfer between the two fluids. The fluids can be either liquids or gases on either the shell or the tube side. For efficient heat transfer, the area for energy exchange should be as large as reasonably possible. This area is directly proportional to tube size, number of tubes and number of passes. By using these heat exchangers (i.e., both the vent condenser and the basement heat exchanger), waste heat can be recovered, thus leading to the conservation of energy. Both of the heat exchangers are Bell & Gosset U-tube SU design type heat exchangers (see Fig. 46). The heat exchanger is termed "SU" as it is designed for steam to flow in the shell, and water to flow in the tubes. This section deals with estimating the basement heat exchanger surface contact area.

Based on the flow direction, heat exchangers can be separated into three categories [18]

1. Parallel flow
   Both fluids flow in the same direction through the heat exchanger.

2. Counter-flow
   The fluids flow in opposite directions in the heat exchanger.

3. Cross-flow
   The fluids flow perpendicular to each other in the heat exchanger.

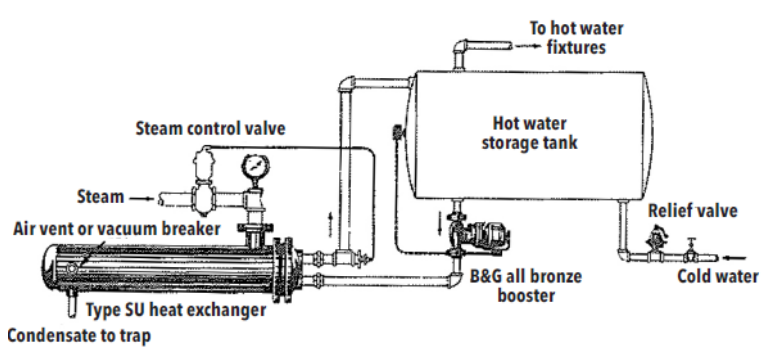


Fig. 46. SU heat exchanger when used with a storage tank (reproduced from Ref. 19)

It is important to know what types of heat exchangers are installed in the power plant because, in the future, similar and compatible heat exchangers can be installed so as to help increase energy savings. There was not sufficient manufacturer's information on the heat exchangers to know if they were parallel flow type or counter-flow type. A very rough calculation was done to estimate the area of the basement

heat exchanger when parallel flow or counter-flow was assumed. These areas were compared to the surface contact area which was estimated from the physical length measurements of the basement heat exchanger (provided by the Bell & Gosset representatives [19] and by this author), to see whether the parallel or counter-flow assumption was correct. The assumed temperature profiles of the makeup and blowdown water flowing through the heat exchanger in the basement are shown in Figs. 47 and 48, with the assumption of parallel flow and counter-flow, respectively.

Temperatures at the inlets and outlets of the heat exchanger were recorded by the HOBO data logger at 4:30:00 pm on December 5, 2015. The temperature of the condensed water from the flash tank was 227 ^{0}F (not steam as pressures were approximately 5-10 psig) while the temperature of the water drained from the heat exchanger to the sewers was 199 ^{0}F. The temperatures of makeup water were 95 ^{0}F at the inlet and 194 ^{0}F at the outlet of the basement heat exchanger. So, temperature rise in the water, ΔT_{rise}, was 99 ^{0}F. The density of water, ρ, was 61 $\frac{lb_m}{ft^3}$ and the makeup water flow rate (that went into the basement heat exchanger),$Q_{vent/HEX}$, was found to be approximately 12 gpm. So the hourly makeup water, $m_{makeup,hr}$, was calculated by using

$$m_{makeup,hr} = \rho\, Q_{vent/HEX} \left(0.00228\ \frac{ft^3}{s\ gpm}\right)(3600\ \frac{s}{hr}) \tag{5a}$$

$$= (61\ \frac{lb_m}{ft^3})\ (12\ gpm)\left(0.00228\ \frac{ft^3}{s\ gpm}\right)\left(3600\ \frac{s}{hr}\right) = 6005\ \frac{lbm}{hr}$$

To calculate the heat gain by the water, use

$$q = m_{makeup,hr}\, c_p\, \Delta T_{rise} \tag{5b}$$

$$= \left(6005\,\frac{lbm}{hr}\right)\left(1.004\,\frac{Btu}{lb_m\,°\mathrm{F}}\right)(99\ °\mathrm{F}) = 590{,}000\ Btu/hr$$

The calculations were done for one specific set of data on December 5, 2015.

From the first law of thermodynamics, the formula for calculating heat transfer occurring in a heat exchanger is [18]

$$q = \eta_{HEX} U A_{calc} \frac{\Delta T'' - \Delta T'}{\ln\frac{\Delta T''}{\Delta T'}} = \eta_{HEX} U A_{calc} \Delta T_{lm} \tag{5c}$$

where q is the heat transfer, U is for the overall heat transfer coefficient, A_{calc} is the contact area for heat exchange to take place, η_{HEX} is the heat exchanger efficiency and ΔT_{lm} is the logarithmic-mean temperature difference. The logarithmic mean temperature, ΔT_{lm}, for a parallel flow heat exchanger is not equal to that for a counter-flow heat exchanger. Using the data available, very approximate calculations were done to estimate the area, A_{calc}, of the heat exchanger; and then this area was compared to the contact area computed from external measurements of the heat exchanger. This could help in deducing whether the heat exchanger is counter-flow type or parallel flow type.

Most of the information below was obtained from contacting Bell & Gosset's technical support group [19].

Overall Heat Transfer Coefficient [19] was,

$$U = 942\frac{Btu}{hr\ °\mathrm{F} ft^2}$$

The diameter of the tubes and the length of the entire heat exchanger was first measured in the power plant using Vernier calipers and a measuring tape, respectively. The outside diameter of the tubes, D, was measured by the author to be 1.98 inches while the overall length of the heat exchanger, L, was measured by the author to be 63 inches. On contacting the Bell & Gosset's representatives, the diameter of the tubes (2 inch), and the length of the heat exchanger (63.375 inches) was provided [19], which agreed very closely with the measurements taken by the author. Also, it was confirmed by company representatives that the heat exchanger had two passes. The exit and entry of the makeup water were on the same side of the heat exchanger.

To compute the area resulting from the measurements, use

$$A = Nn\pi DL \tag{6}$$

where N is the number of tubes and n is the number of passes. For two tubes, two passes, and using D and L provided by the company representatives,

$$A = (2)(2)\pi(2\ in)(65.375\ in) = 1643\ in^2 = 11.41\ ft^2$$

The diameter of the heat exchanger (without insulation) was measured by the author to be 8 inches. It can be assumed that the maximum number of tubes that can be contained in the heat exchanger is four (i.e., 8 inches/2 inches = 4). The heat exchanger does not have enough area and space to contain more than four tubes. Assuming that there are four tubes (instead of two), the external surface tube area is computed as (from Eq. (6))

$$A = (4)(2)\pi(2\ in)(65.375\ in) = 3286\ in^2 = 22.82\ ft^2$$

If parallel flow is assumed, then more tubes (i.e., three) would be required to see whether the external surface tube area could match the calculated area. The external surface tube area for 3 tubes is (from Eq. (6))

$$A = (3)(2)\pi(2\ in)(65.375\ in) = 2464\ in^2 = 17.11\ ft^2$$

Next, assumptions of parallel flow and counter-flow will be used to compute areas to compare with the measured area.

3.iv.a Parallel Flow Assumption

Physically, the heat exchanger piping structure is such that it looks like it is counter-flow. However, in this section, parallel flow type was assumed so as to see whether the calculations for the area matched the measured surface tube area. Thus, the directions of flow of the hot water and the cold makeup water into the heat exchanger were assumed to be the same (refer to Fig. 47).

At the inlet of the heat exchanger, the temperature of hot water entering the heat exchanger (from the flash tank), T_{h1}, was 227 ^{0}F, and the temperature of cold makeup water entering the heat exchanger, T_{c1}, was 95 ^{0}F. These temperatures were obtained from the digital TMC6-HE temperature sensors (see Appendix B1) at the inlet pipe lines for the basement heat exchanger.

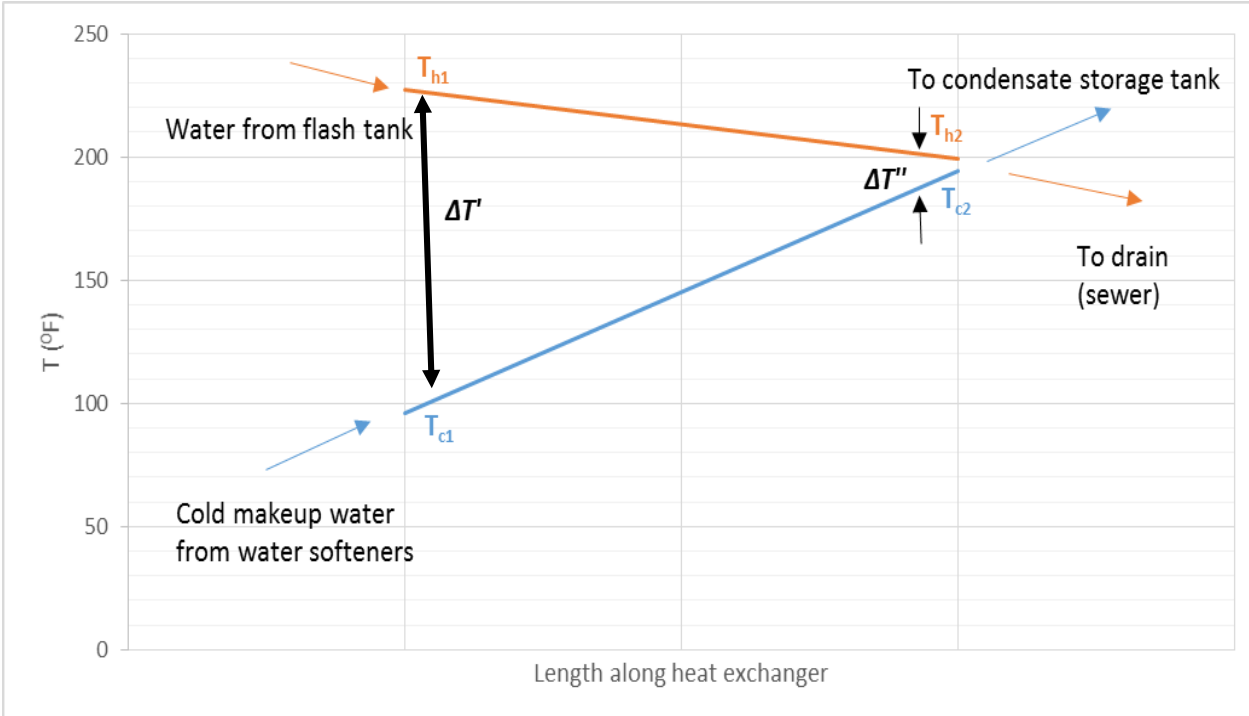


Fig. 47. Temperature profile of the SU heat exchanger (parallel flow)

At the outlet of the heat exchanger, the temperature of the hot water drained to the sewer, T_{h2}, was 199 ^{0}F, and the temperature of heated makeup water leaving the heat exchanger, T_{c2}, was 194 ^{0}F. These temperatures were obtained from the digital temperature sensors at the outlet pipelines for the basement heat exchanger.

Therefore, the temperature difference at the heat exchanger exit, ΔT'', was 199 – 194 ^{0}F or 5 ^{0}F. The temperature difference at the heat exchanger inlet, ΔT', was 227 – 95 ^{0}F, or 132 ^{0}F. From the known values of ΔT'' and ΔT', the calculated ΔT_{lm} was 38 ^{0}F. So, using Eqs. (5b) and (5c), the calculated heat exchanger area of the tubing, A_{calc}, was 16.48 ft^2. η_{HEX} was assumed to be 1 for this calculation.

3.iv.b Counter-Flow Assumption

For this calculation, the basement heat exchanger was assumed to be a counter-flow design; and, the directions of flow of the hot water, from the flash tank and the makeup water from the water softeners, were assumed to be opposite (refer to Fig. 48).

T_{h1}, T_{h2}, T_{c1} and T_{c2} were the same as given in Section 3.iv.a for parallel flow.

Thus, ΔT'' was 227 ^{0}F – 194 ^{0}F or 33 ^{0}F. ΔT' was 199 ^{0}F – 95 ^{0}F or 104 ^{0}F. From the known values of ΔT'' and ΔT', the calculated ΔT_{lm} was 63.1 ^{0}F. So, using Eqs. (5b) and (5c), calculated area, A_{calc}, was 9.92 ft^2. η_{HEX} was assumed to be 1 for this calculation.

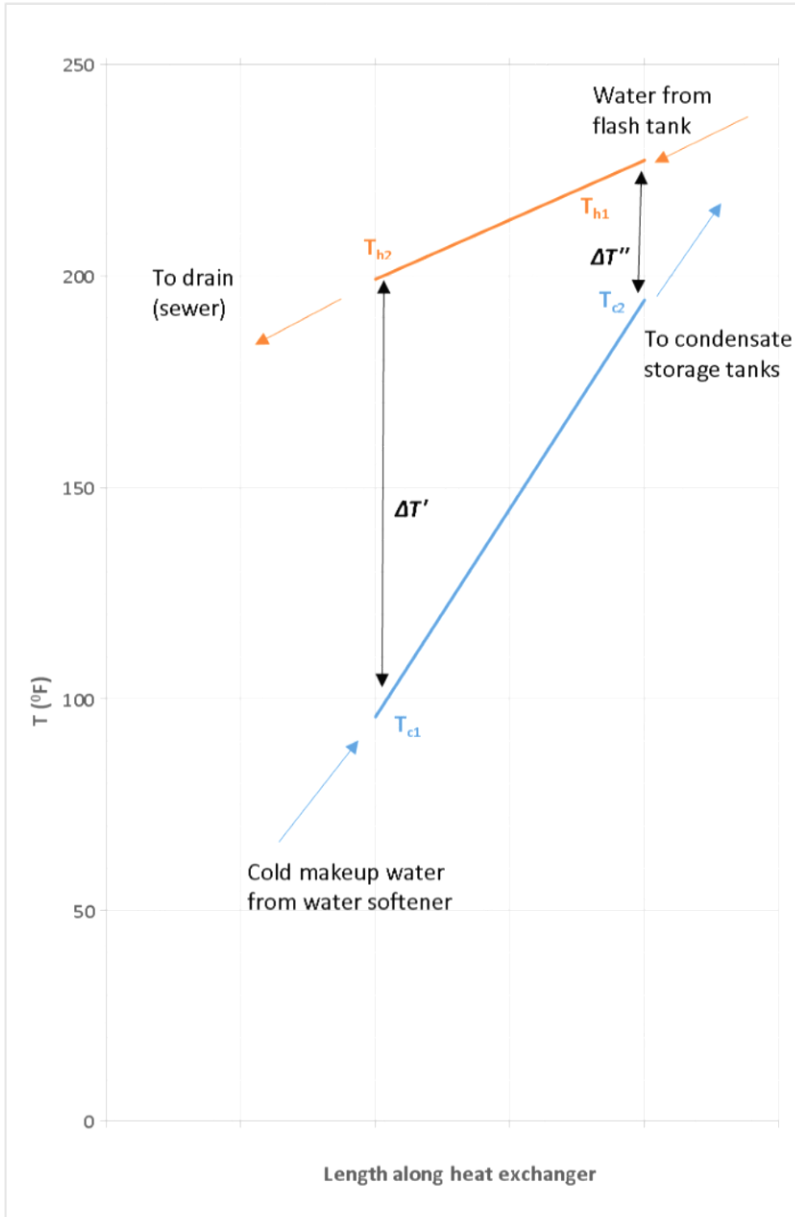


Fig. 48. Temperature profile of the SU heat exchanger (counter-flow)

The type of heat exchanger could best be determined if more information, such as η_{HEX}, from the manufacturer was available. The efficiency of the SU heat exchanger is unknown (the Bell & Gosset representatives couldn't supply this information). Typically, heat exchanger efficiencies vary between 0.85 and 0.9 [18]. Also, there is flow of water in both the shell and the tube sides of the basement heat exchanger. But the heat exchanger (being SU type) was designed for flow of steam on the shell side [19].

Table 3 shows computed A_{calc} and *A* values when either parallel flow or counter-flow was assumed for the heat exchanger in the basement.

$A_{calc}$was found by assuming a η_{HEX} of 1 and 0.9, respectively (see Table 3). Also, *A* was calculated from measurements and then assuming N of 2, 3 and 4 tubes (see Table 3).

Table 3. Calculated and measured heat exchanger areas based on parallel or counter-flow assumption

	Parallel Flow	**Counter-Flow**	**Measured**
$\boldsymbol{\Delta T_{lm}}$ **(^{0}F)**	**38**	**63**	
$\boldsymbol{A_{calc}}$ **(ft^2) (for $\boldsymbol{\eta_{HEX} = 1}$)**	**16.48**	**9.92**	
$\boldsymbol{A_{calc}}$ **(ft^2) (for $\boldsymbol{\eta_{HEX} = 0.9}$)**	**18.31**	**11.02**	
***A* (ft^2) (2 tubes)**			**11.41**
***A* (ft^2) (3 tubes)**			**17.11**
***A* (ft^2) (4 tubes)**			**22.82**

For a η_{HEX} of 0.9, it can be observed that the counter-flow assumption for two tubes (i.e., N=2) yields a measured external tube surface area (A) close to the calculated external tube surface area (A_{calc}) of the heat exchanger, as compared to that of the parallel flow assumption for three tubes (i.e., N=3) (see Table 3). So, it can be suggested that the heat exchanger is counter-flow if the assumptions of $\eta_{HEX} = 0.9$ and N = 2 made in this section are correct.

Chapter 4: Analysis of Pump Data

This chapter deals with the data collected from the Worthington constant speed pump (pump #1) and the Grundfos variable speed pumps (pump #2). As explained in Chapter 1, the power plant staff have four condensate pumps, and each pump runs for one week out of every four weeks. Pump data was collected from June of 2015 to November of 2015. This data was used to calculate and compare the power consumption of the Worthington and Grundfos pumps. The level control mode versus pressure control mode of the Grundfos pumps was also investigated to see its effect on power consumption.

For a specified impeller diameter and speed, a centrifugal pump has a fixed and predictable performance curve (although with age, changes do occur on the performance curve) (see Appendix H for some examples of pump performance curves). The "pump curve" shows the ability of the pump to produce a certain flow rate based on the pump head (discharge pressure) at that operating condition. The point where the pump operates on its curve is dependent upon the characteristics of the system in which it is operating, and these characteristics can be translated into the "system curve". The system curve is the relationship between fluid flow and hydraulic losses in a system. The parabolic shape of the system curve is determined by the frictional losses (dependent on the square of the flow rate) through the system, including all pipe lengths, bends and valves. The KU plant has control valves and other mechanical valves, mostly in series. For cases considered in this thesis, the system curve is dependent primarily on the opening/closing of the control valve before the DA tank. The operating point is at the intersection of the system curve and pump curve (see Fig. 49). The opening/closing of this valve changes the system curve, which then determines whether the operating point moves to the right or the left in Fig. 49.

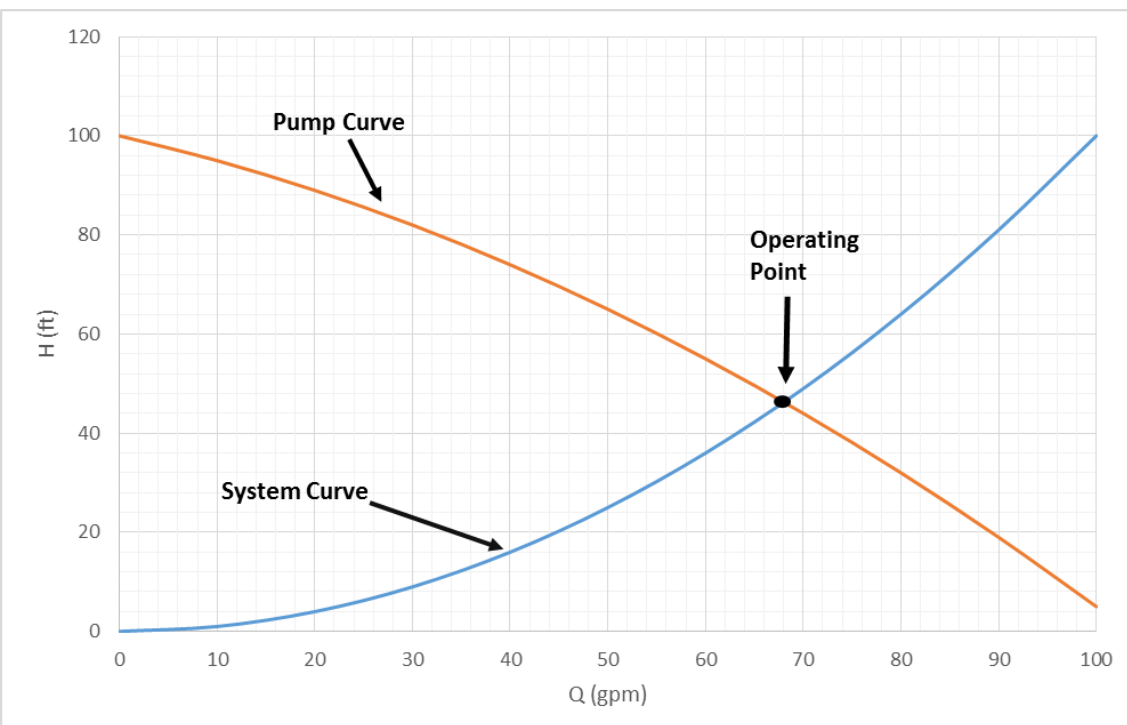


Fig. 49. Example of pump curve, system curve and operating point

4.i Data from Worthington Constant Speed Pump

The Worthington constant speed pump ran during the week from June 29 through July 6 of 2015. The discharge pressure of the Worthington pump varied because it depended on the pump's operating point, which itself was affected by system curve changes (see Fig. 50). The average discharge pressure was found to be 48.46 psig. The system curve changed because the control valve opened and closed in order to keep the water level in the DA tank fairly constant (i.e., 52% of tank capacity). The control valve wasted energy because it blocked water flow into the DA tank. Thus, the system curve's shape changed primarily due to the frictional losses generated by the control valve. When the control valve opened more, this caused the system curve to become flatter. The operating point then moved to the right in Fig. 49, resulting in increased flow of water into the DA tank. When the control valve closed more, and the system curve became steeper; and the operating point moves to the left, resulting in decreased flow of water into the DA tank.

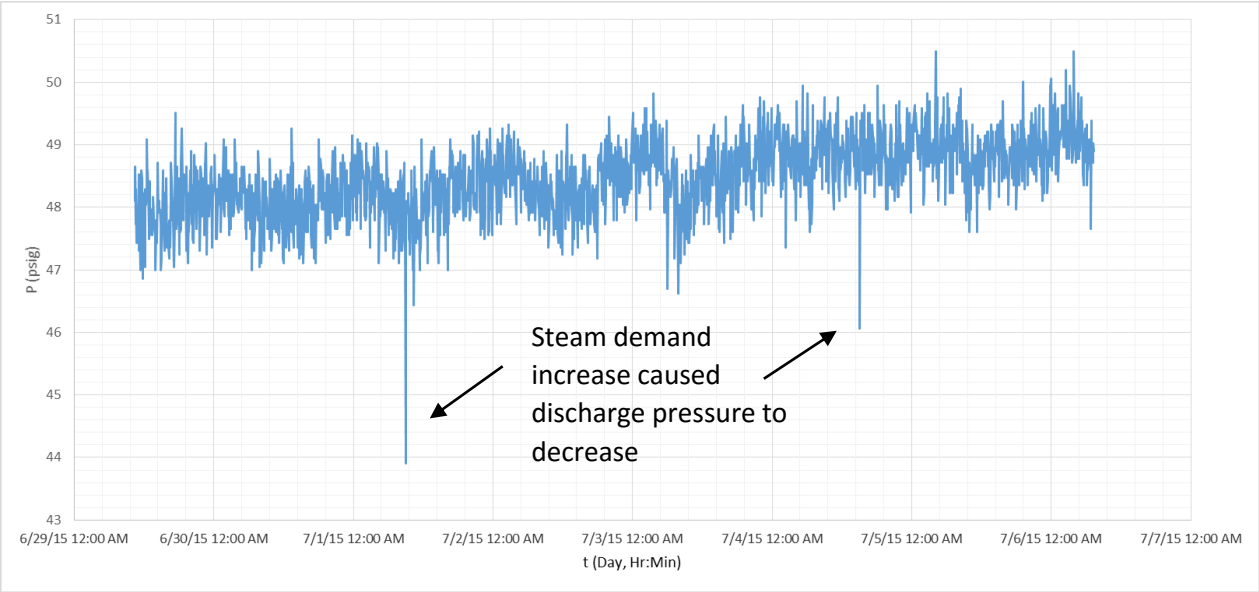


Fig. 50. Discharge pressure of Worthington pump when it ran from June 29 to July 6 of 2015

As explained before, the Worthington pumps only ran for one week in each month during the time period of the study. The average discharge pressures for the Worthington pump during one week for each month from June 2015 to November 2015 are presented in Table 4.

When the steam demand increased, the control valve opened more in order to feed the DA tank with more condensate water. So the constant speed pump tried to deliver a higher flow rate; but the discharge pressure reduced due to the fact that this pump could not perform more work than that for which it was designed (see Fig. G1a for Worthington pump curves). The average flow rate of the Worthington constant speed pump was found to be 136.58 gpm, during the week of June 29 to July 6 of 2015 (see Fig. 51).

Table 4. Average discharge pressure of the Worthington pump between June and November of 2015 (for the six weeks when the pump ran)

Month	Discharge Pressure (psig)	Avg. Pressure at point just before Control Valve (psig)	Avg. Differential Pressure across Control Valve (psig)	Avg. Differential Pressure from Pump to Control Valve (psig
June	47.72	39.44	30.28	8.28
July	48.46	39.62	29.77	8.84
August	49.23	39.61	30.12	9.62
September	48.81	39.13	31.49	9.68
October	49.02	39.19	32.81	9.83
November	49.06	35.93	30.32	13.13
Average	48.71	38.82	30.79	9.89

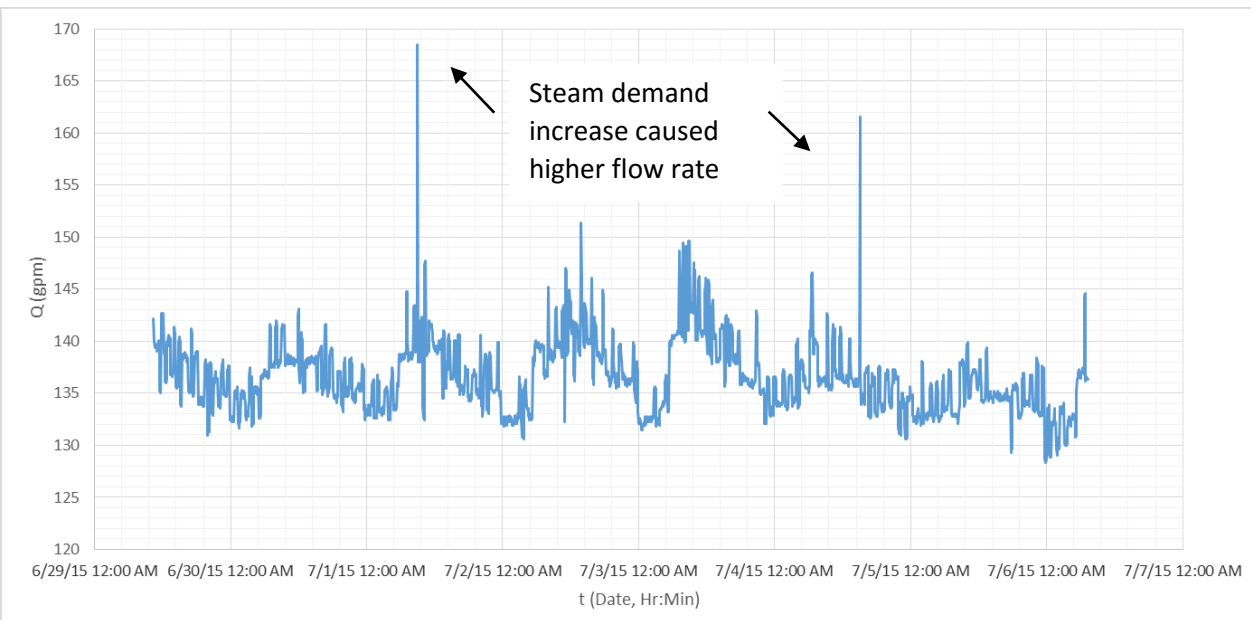


Fig. 51. Discharge flow rate of Worthington pump when it ran from June 29 to July 6 of 2015

The pressure before the control valve, during the week of June 29 to July 6 of 2015, is presented in Fig. 52. This data was taken in order to determine the pressure drop in the pipelines between the Worthington pump discharge and a point just before the control valve. There are numerous pipe fittings such as elbows, valves and T-sections in these pipelines. Minor head losses occur because of the restriction in flow by these pipe fittings. Also, there are major head losses due to friction in the pipelines. Because of these major and minor head losses, pressure drop is created as the water moves inside the pipelines. During the week of June 29 to July 6 of 2015, the average pressure before the control valve was 39.62 psig (see Table

4), while the discharge pressure of the Worthington pump was 48.46 psig (see Table 4). Thus, there was a pressure drop (ΔP) of approximately 8-10 psig (see Table 4) from the discharge of the Worthington pump to the control valve, because of the major and minor head losses.

Next, Fig. 53 shows the excess condensate water flow rate to the vent condenser, when the Worthington pump ran during the week of June 29 to July 6 of 2015. The condensate water split up, and most of the water went to the vent condenser while the rest of the water went to the DA tank. The discharge pressure of the Worthington pump (i.e., 48.46 psig) provided enough pressure for the water to reach the vent condenser on the first floor (an approximate 40 ft of elevation which is equivalent to 17.31 psig). The average flow rate to the vent condenser during the week of June 29 to July 6 of 2015 was 92.92 gpm (see Table 5).

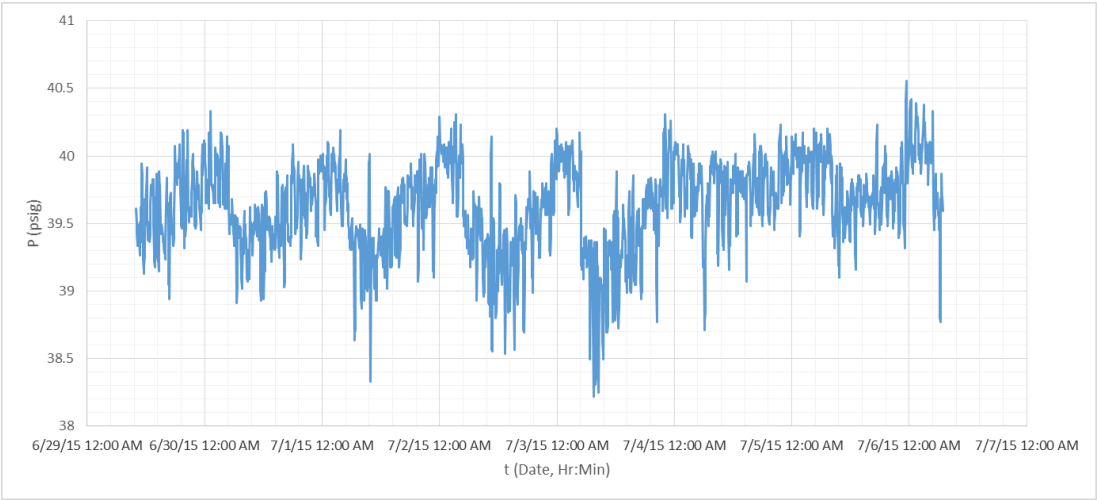


Fig. 52. Pressure before the control valve when Worthington pump ran from June 29 to July 6 of 2015

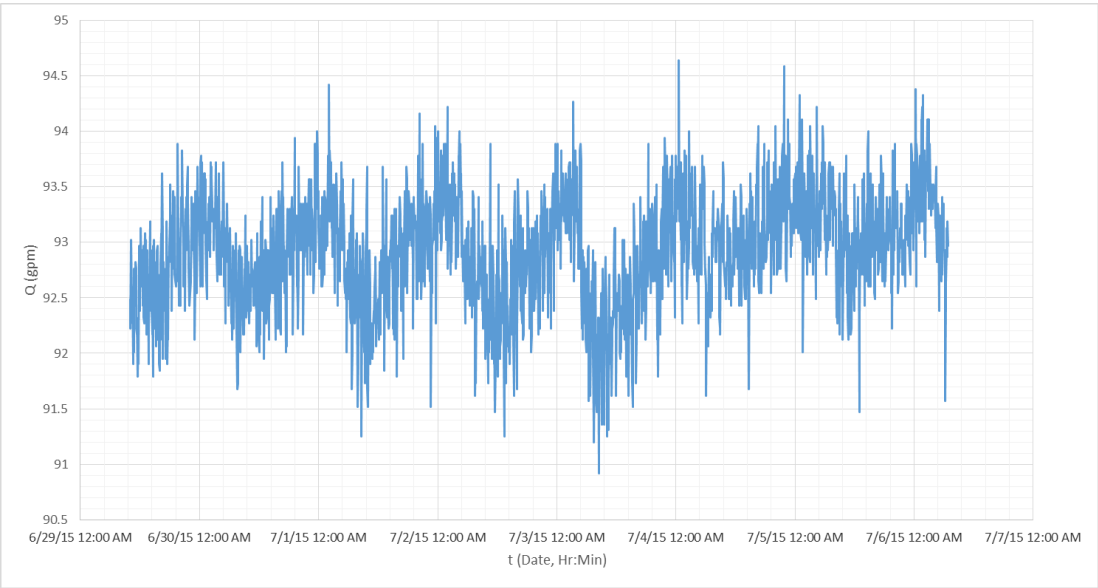


Fig. 53. Flow rate of condensate water to the vent condenser when Worthington pump ran from June 29 to July 6 of 2015

Table 5 shows the average flow rates of the condensate water (i.e., the pump discharge flow rate, the excess condensate flow rate to the vent condenser, and the flow rate of condensate to the DA tank) during the six weeks when the Worthington pump was running from June of 2015 to November of 2015.

Table 5. Average flow rates of the condensate water for the six weeks when the Worthington pump ran between June and November of 2015

Month	Avg. Pump Discharge Flow Rate (gpm)	Avg. Flow Rate to Vent Condenser (gpm)	Avg. Flow Rate to DA Tank (gpm)
June	139.64	92.23	47.41
July	136.58	92.92	43.66
August	136.74	92.82	43.92
September	142.97	92.13	50.84
October	135.81	92.11	43.70
November	182.75	88.78	93.97
Average	145.76	91.83	53.91

The recorded average power consumption of the Worthington pump was 5.18 kW (see Fig. 54), during the week of June 29 to July 6 of 2015. This power was recorded directly from the Veris power monitor (see Appendix B8), which was connected to the HOBO data logger (see Appendix B3). The power consumed by the Worthington pump followed the same trend as the discharge flow rate (see Fig. 51). This is because, with an increase/decrease in steam demand, there was decrease/increase in pressure in the DA tank. This led to more/less water discharged by the pump to the DA tank and vent condenser and hence, the pump needed to work more/less for the increased/decreased condensate flow rate. The discharge pressure of the pump thus decreased/increased to compensate for the increased/decreased condensate flow.

Two methods were used to check the validity of the recorded power consumption. The first approach was reading the power consumption from the pump manufacturer's performance curves (see Figs. G1a and G2a for Worthington and Grundfos pump curves respectively). The second approach was employing the standard equation to calculate the power using measured pump pressure, measured flow rate, and Fig. G1a or Fig. G2a efficiency. All the calculations for the Grundfos pumps have been done in Section 4.ii.

In the first approach, for specifying the average operating point, one piece of information is needed for the Worthington pump curve (see Fig. G1a): the average discharge flow rate of the pump. For the Grundfos pump curve (see Fig. G2a), the average discharge pressure is also needed. The average discharge pressure is shown on the Grundfos pump curve as pressure head in feet instead of psig. Therefore, the average recorded discharge pressure given in psig had to be converted to feet. This information is used in Section 4.ii. The term head can be defined as "the quantity used to express the energy content of the

liquid per unit weight of the liquid referred to any arbitrary datum" [20]. To convert the pressure from psig into feet of head, use [20]

$$H = P\ \frac{2.31(\frac{in^2}{lb_f}ft)}{\gamma'} \tag{7}$$

All terms are defined in the Nomenclature. The specific gravity (γ') for water at 160 ^{0}F [7] is 0.979 [20] (the average temperature of the condensate water was assumed to be 160 ^{0}F). Using the pump's average discharge flow rate (in gpm), the Worthington pump performance curves were used (see Fig. G1a) in order to find the required power to perform the work for the Worthington pump. Using the pump's average discharge flow rate (in gpm) and discharge pressure (in ft), the Grundfos pump performance curves were used (see Fig. G2a) in order to find the required power to perform the work for the Grundfos pumps in Section 4.ii.

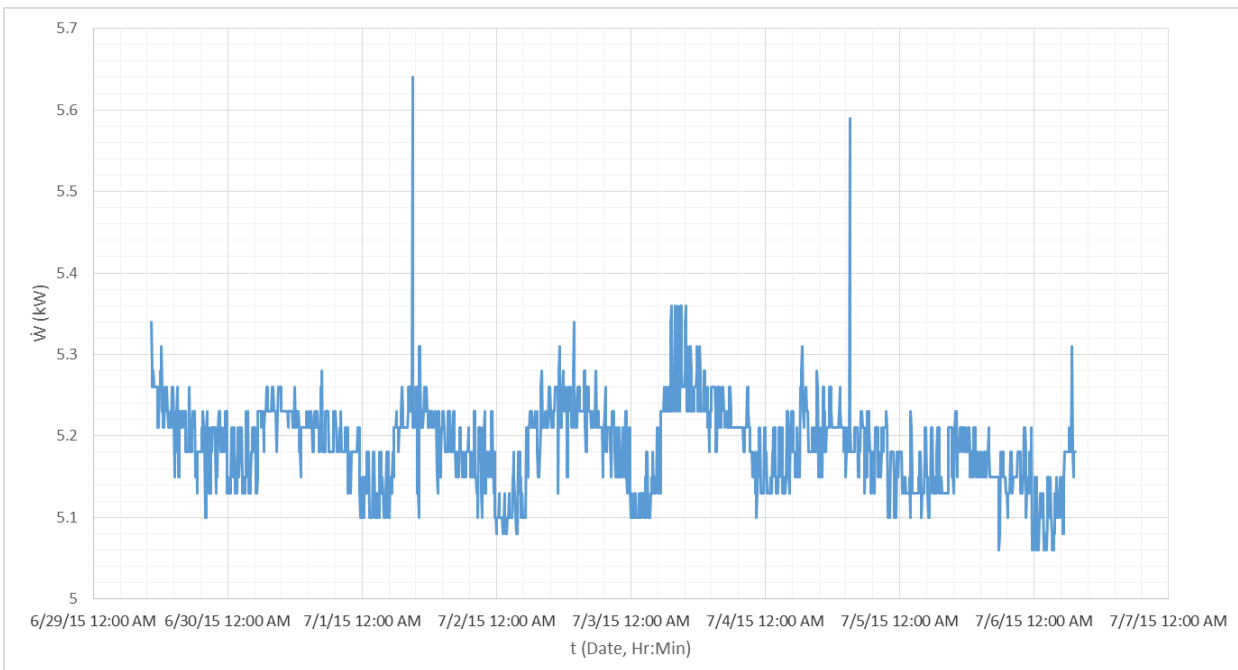


Fig. 54. Power consumed by Worthington pump when it ran from June 29 to July 6 of 2015

The second approach for validating the recorded power consumption used measured flow rate, measured pressure and efficiency (from Fig. G1a) to calculate the brake horsepower for the pump [20].

$$bhp = \frac{Q\ \Delta P}{C_1\ \eta_p} \tag{8}$$

However, in order to calculate the electrical power consumption by the system (pump and motor), the motor and pump efficiencies were required. Therefore, for the Worthington pump, Eq. (8) becomes

$$\dot{W}_2 = \frac{Q\ \Delta P}{C_1\ (\eta_m)(\eta_p)}\ (0.7456\ \frac{kW}{hp}) \tag{9a}$$

and for the Grundfos pumps, Eq. (8) becomes

$$\dot{W}_2 = \frac{Q\ \Delta P}{C_1\ \ (\eta_m)(\eta_V)(\eta_p)}\ \ (0.7456\ \frac{kW}{hp}) \tag{9b}$$

All terms are defined in the Nomenclature. Equation (9b) is used in Section 4.ii.

The percent difference between the power from the pump curve and the power recorded by the HOBO data logger was computed. Also, the percent difference between the computed power from Eq. (9a) (or Eq. (9b), depending on whether the pump was Worthington or Grundfos) and the power recorded by the HOBO data logger was computed. The following equations were used.

$$E\ (\%) = \frac{\dot{W}_1 - \dot{W}}{\dot{W}} \times 100 \tag{10}$$

$$E'(\%) = \frac{\dot{W}_2 - \dot{W}}{\dot{W}} \times 100 \tag{11}$$

$$E''(\%) = \frac{\dot{W}_2 - \dot{W}_1}{\dot{W}_1} \times 100 \tag{12}$$

Here, $\dot{W}$ is the power recorded by the HOBO data logger, $\dot{W}_1$ is the power read from the pump curve, and $\dot{W}_2$ is the power computed from Eq. (9a) (or Eq. (9b), depending on whether the pump was Worthington or Grundfos).

During the week of June 29 to July 6 of 2015, the Worthington pump's power consumption from pump curve (see Fig. G1a) was approximately 4.49 kW, using an average flow rate of 136.58 gpm. The average recorded power consumption from the HOBO data logger was 5.18 kW, 13.32% higher than the power from the pump curve.

For Eq. (9a), the pump efficiency must be found. Using an average flow rate of 136.58 gpm for the Worthington pump, the efficiency from the pump curve was found to be 0.59 (see Fig G1a), and the pump power consumption was then computed to be 4.88 kW, with η_m taken to be 1.0. It is important to note that the power read from the pump performance curve for the Worthington pump depended only on the pump's discharge flow rate. The calculated power consumption from Eq. (9a) was 5.79% less than the recorded power consumption value of 5.18 kW.

Table 6 shows the average power and the computed power differences for the Worthington pump for each of the months from June of 2015 to November of 2015. The table shows that the differences are consistent throughout those months. The Worthington pump was installed in the power plant in 2005, while the power was recorded by the data logger in 2015. Thus, aging of the pumps is assumed to have caused deterioration in the pump performance, and hence a decrease in the efficiency.

The aging factor contributes to calculation errors because the pump manufacturer has no information regarding aging pump performance. The efficiency of the pump reduces gradually with age due to wear and tear. So, the pump's power consumption increases. Also, there was no electronic version of the Worthington pump curve. So, the pump power and efficiency from the pump curve were determined by hand. An error of ±5% had to be considered for the power read from the pump curve.

The power read from the pump curve varied greatly as compared to that of the calculated power using Eq. (9a). It has to be noted that there was no information about the type of efficiency on the pump performance curve; whether it was the overall efficiency or just the motor efficiency. If it was just the motor efficiency, then the pump efficiency also had to be taken into account. Multiplying both the motor and pump efficiencies would give the overall efficiency. Then the power calculated from Eq. (9a) could become more accurate and might have a value closer to the power read from the pump curve. In this thesis, the pump curve efficiency for the Worthington pump was assumed to be overall efficiency because of lack of information.

Table 6. Average power determined from three sources and comparison of those results for the Worthington pump

Month (2015)	June	July	August	September	October	November
Recorded Power (kW) (from HOBO Data Logger)	5.12	5.18	5.17	5.30	5.21	6.03
Power Read from Pump Curve (kW)	4.54	4.49	4.51	4.62	4.47	5.17
Power Calculated from Eq. (9a) (kW)	4.99	4.88	4.88	5.11	4.85	5.70
E (%) (Difference between Recorded Power and Power Read from Pump Curve)	-11.32	-13.32	-12.76	-12.82	-14.20	-13.12
E' (%) (Difference between Recorded Power and Power Calculated from Eq. (9a))	-2.53	-5.79	-5.46	-3.57	-6.88	-5.47
E'' (%) (Difference between Power Read from Pump Curve and Calculated Power from Eq. (9a))	9.91	8.68	8.20	10.60	8.50	10.25

4.ii Data from Grundfos Variable Speed Pumps

The Grundfos pumps ran at constant pressure during the week of October 27 to November 2 of 2015, apart from two hours on October 29, when they ran in level control mode. The Grundfos pumps also ran in level control mode for two hours on October 21, 2015.

The operating principle of the Grundfos pump was different from that of the Worthington pump. Whenever the operating point of the pump changed due to opening/closing of the control valve (to keep the water level in the DA tank constant), the variable speed pumps sped up or slowed down to maintain the set point discharge pressure. The set point for the discharge pressure was selected by the steam power plant's staff to be 43 psig (see Fig. 55). When the steam demand increased, the variable speed pumps responded to the increase and provided a higher flow rate at the same discharge pressure of 43 psig (see Fig. 56). For Fig. 56, the average pump discharge flow rate was found to be 175 gpm from the Grundfos

control panel. The corresponding average pump discharge flow rate from the Siemens flow meter was 150 gpm.

There was a relatively consistent difference of approximately 24 gpm between the flow rate data obtained from the Siemens flow meter and the Grundfos control panel. Table 7 shows the average flow rates of the Grundfos pumps, obtained from both the control panel and the Siemens flow meter. The differences were consistent throughout all of the months, as can be seen in Table 7. On contacting the Grundfos representatives [21], it was confirmed that the control panel data should be trusted. However, the Grundfos control panel does not have a built-in flow meter, so it does not measure flow rate of the pumps.

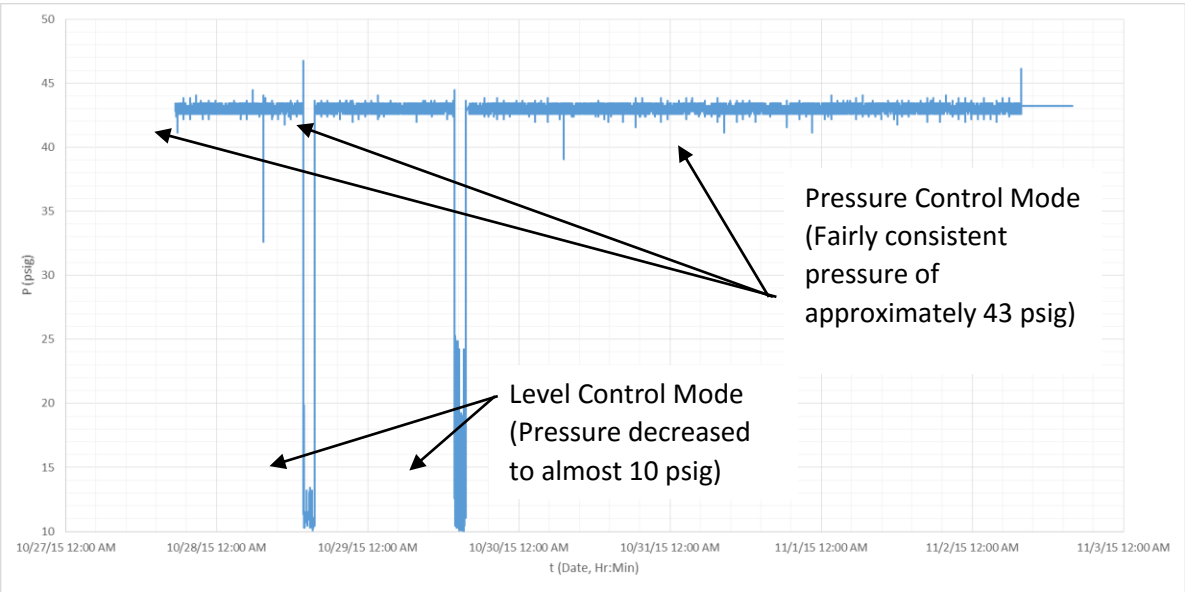


Fig. 55. Discharge pressure of Grundfos pumps from October 27 to November 2 of 2015

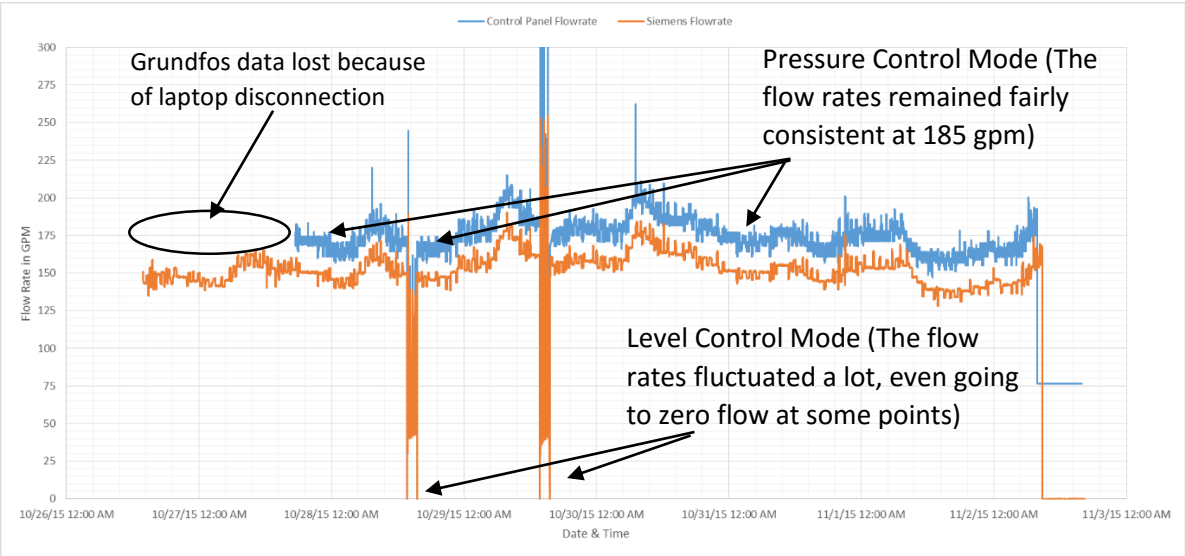


Fig. 56. Flow rate of Grundfos pumps from October 26 to November 2 of 2015

Only the power consumption and the discharge pressure are measured. Based on this data, the Grundfos control panel then calculates the discharge flow rate. Instead, the Siemens flow meter measures the actual discharge flow rate of the pumps. The flow rate data from the Siemens flow meter was trusted because the flow meter had been tested and calibrated [1]. For these reasons, the control panel data for flow rate was not considered for calculations (for power consumption of the Grundfos pumps). All of the calculations henceforth use the flow rate data from the Siemens flow meter.

Table 7. Flow rate comparison between Siemens flow meter and Grundfos control panel for six weeks of data in the months when the Grundfos pumps ran

Flow rate (Q)	June	July	August	September	October	November
Control Panel (gpm)	**163.86**	**166.42**	**166.25**	**164.13**	**175.22**	**174**
Siemens Flow meter (gpm)	**137.83**	**142.44**	**142.59**	**140.69**	**149.63**	**150.72**
Difference between Readings (gpm)	**26.03**	**23.98**	**23.66**	**23.44**	**25.59**	**23.28**

The Grundfos control panel did not record flow rate, discharge pressure and power consumption data for the Grundfos pumps at equal time intervals. Thus, the average flow rates had to be computed based on time-weighting by using the Trapezoidal formula (see Eq. (1) and Section 2.iii). Table 8 shows the differences between the values for the arithmetic average and the time-weighted average for the flow rates.

Table 8. Comparison of arithmetic average and time-weighted average for flow rates of condensate water when the Grundfos pumps ran

Flow Rate (Q) \ Month	June	July	August	September	October	November
Arithmetic Average (gpm)	**164.04**	**166.41**	**166.15**	**164.20**	**174.01**	**174.22**
Time-Weighted Average (gpm) (from Eq. (1))	**163.86**	**166.42**	**166.25**	**164.13**	**175.22**	**174.00**

The time-weighted average of recorded power consumption from the Grundfos control panel was 5.6 kW during the week of October 27 to November 2 of 2015 (see Fig. 57). This was similar to the power consumption of the Worthington pump. This was because the Grundfos pumps usually ran in pressure control mode, i.e., the discharge pressure for the Grundfos pumps were set at a constant pressure of 43 psig. From the next sections, 4.ii.a and 4.ii.b, it can be seen that the power consumption reduced drastically when the Grundfos pumps were run in level control mode.

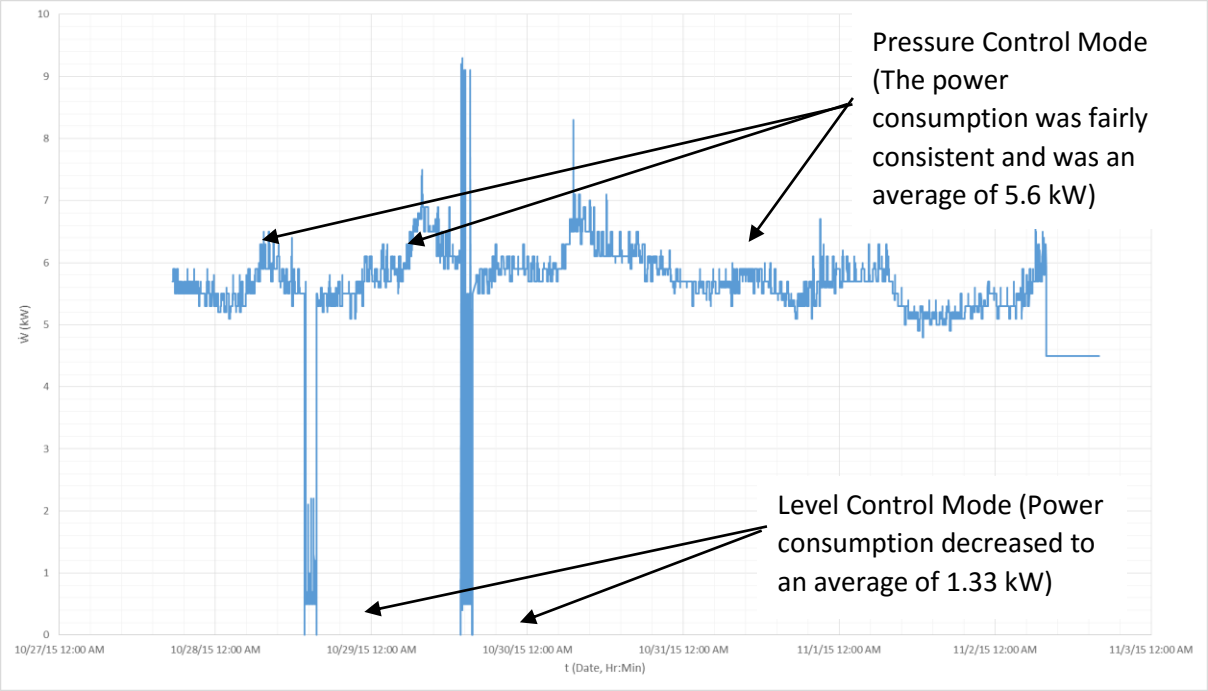


Fig. 57. Power consumption of Grundfos pumps from October 27 to November 2 of 2015

4.ii.a Level Control Mode (October 21, 2015)

Level control mode was successfully implemented on October 21, 2015. This mode was implemented for two hours from 1:30 pm to 3:30 pm. In the following set of results, the Grundfos pumps ran in level control mode, based on the input that the Grundfos control panel received from the level sensor installed at the DA tank (refer to Section 2.ii.b.ii.b). In this operating mode, the Grundfos pumps were unable to deliver the excess condensate water to the vent condenser. Thus, all of the condensate water flowed into the DA tank. The control valve just before the pump was fully open during level control mode, so as to allow for maximum condensate flow into the DA tank.

The power consumption of the Grundfos pumps when running in level control mode vs. pressure control mode is shown in Fig. 58. The power consumption fluctuated when the Grundfos pumps were in level control mode, from 1:30 pm to 3:30 pm on October 21, 2015. The power consumption peaked three times due to the pumps' sensitivity to changes in the water level in the DA tank.

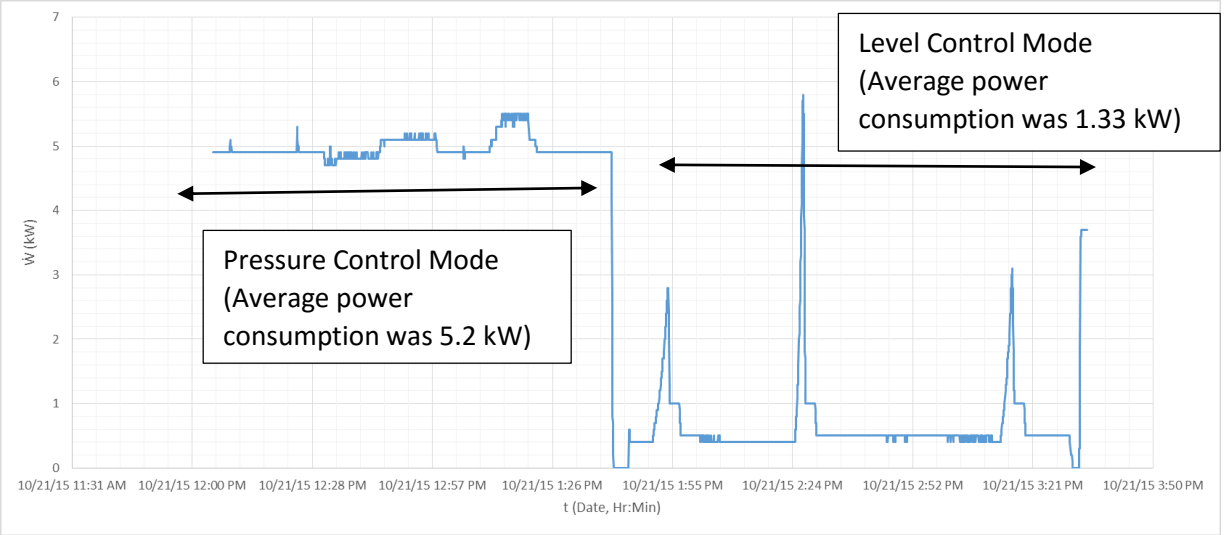


Fig. 58. Power consumption of Grundfos pumps when they ran in pressure and level control modes from 12:00 pm to 3:30 pm on October 21, 2015

Once the water level went below the set point of 52% in the DA tank, the pumps reacted as fast as possible to make up for that reduced water level. Such fast reaction caused the power consumption to peak as shown in Fig. 58. However, these peaks lasted for a short period of time, ranging from 9 to 40 seconds, depending upon the amount of water in the DA tank. Then power consumption dropped steeply to the lowest level. Even though these peaks existed, the Grundfos pumps consumed much less energy during level control mode than when they were in pressure control mode. The average power consumption was only 1.33 kW, which is much lower than the power consumption (approximately 5.2 kW) when the Grundfos pumps were in pressure control mode from 12 pm to 1:30 pm as shown in Fig. 58. The average discharge pressure during level control mode was only 12.94 psig (see Fig. 59), while the average discharge flow rate was only 64.08 gpm (see Fig. 60).

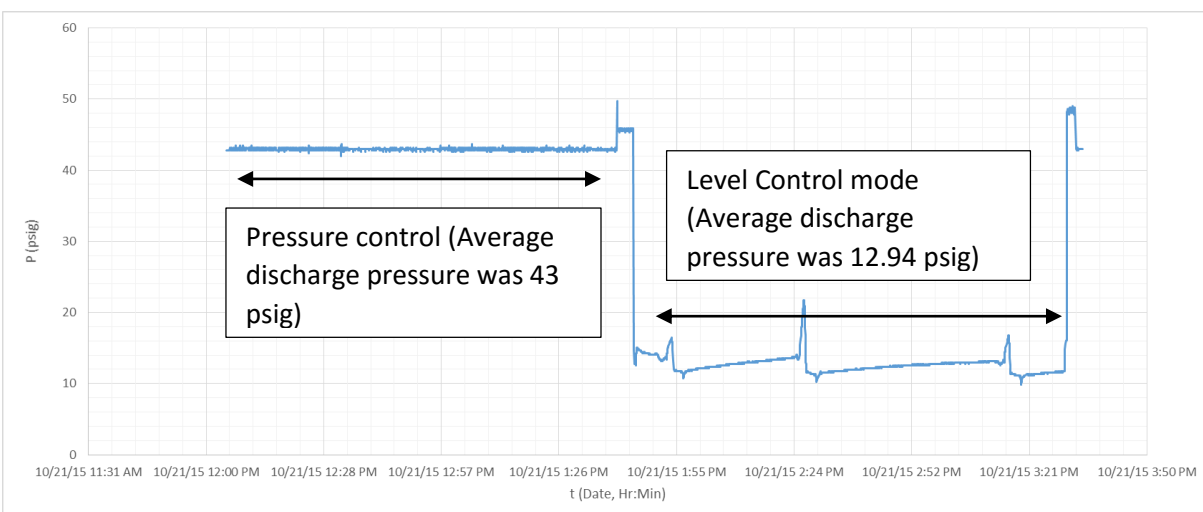


Fig. 59. Discharge pressure of Grundfos pumps when they ran in pressure and level control modes from 12:00 pm to 3:30 pm on October 21, 2015

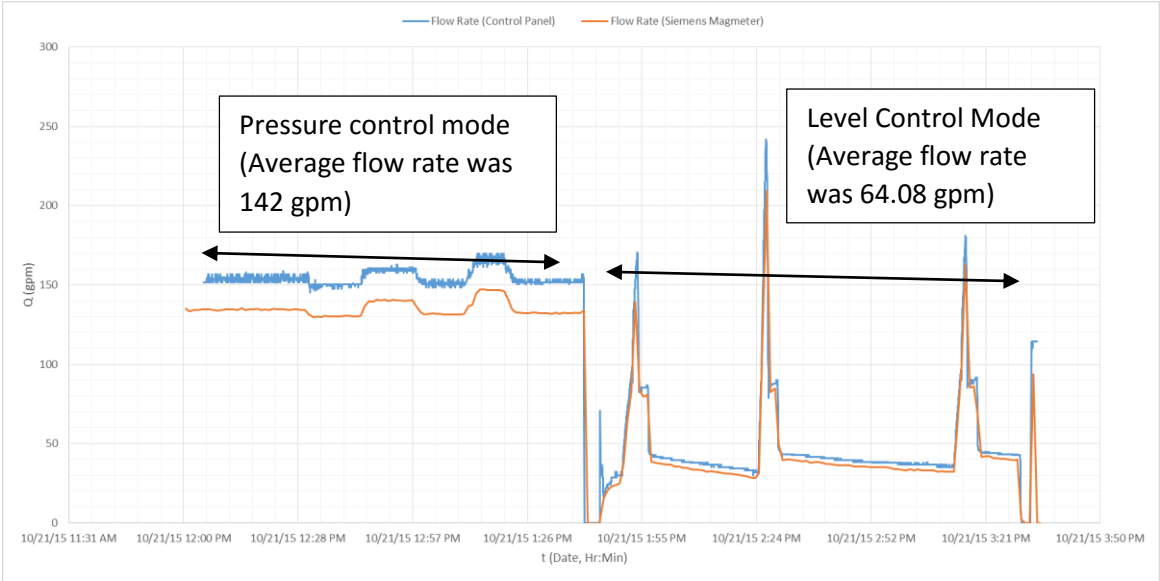


Fig. 60. Flow rate of Grundfos pumps when they ran in pressure and level control modes from 12:00 pm to 3:30 pm on October 21, 2015

4.ii.b Level Control Mode (October 29, 2015)

This section has the details of the level control mode on October 29, 2015 (level control mode was also used on October 28, 2015). The flow rate, discharge pressure and discharge flow rate were also analyzed to see if the data followed similar trends to that of the data recorded for the level control mode period on October 21, 2015. The same settings were selected as those of October 21, 2015, and the test was carried out from 1:30 pm to 3:30 pm.

Also, Ti (the pump response time interval) was selected as 2 seconds, instead of the default 0.5 second. This meant that the Grundfos pumps allowed 2 seconds to pass before they responded to any change in the water level in the DA tank. 0.5 second was deemed too fast (see Fig. 61), and there were many peaks and troughs in the plots when 0.5 second was used. For a Ti of 2 seconds, the peaks and troughs weren't completely eliminated (see Fig. 62), but they were greatly reduced. There were fourteen power consumption peaks for a Ti of 0.5 second, while there were only seven power consumption peaks for a Ti of 2 seconds, and the Grundfos pumps ran in level control mode for two hours in each case. The minimum pump performance was also set at 42% of full speed for both cases (see Section 2.ii.b.ii.b).

A Ti of 2.5 or even 3 seconds is recommended. This could result in fewer peaks and troughs in the plot for pump power consumption, i.e., fewer fluctuations in the pumps' power consumption. However, there is also a possibility that there could be an overflow of water into the DA tank because of the delayed response by the pump to the DA tank's water level. So, any change in the Ti setting has to be carefully made so that there are no adverse effects on the DA tank water level.

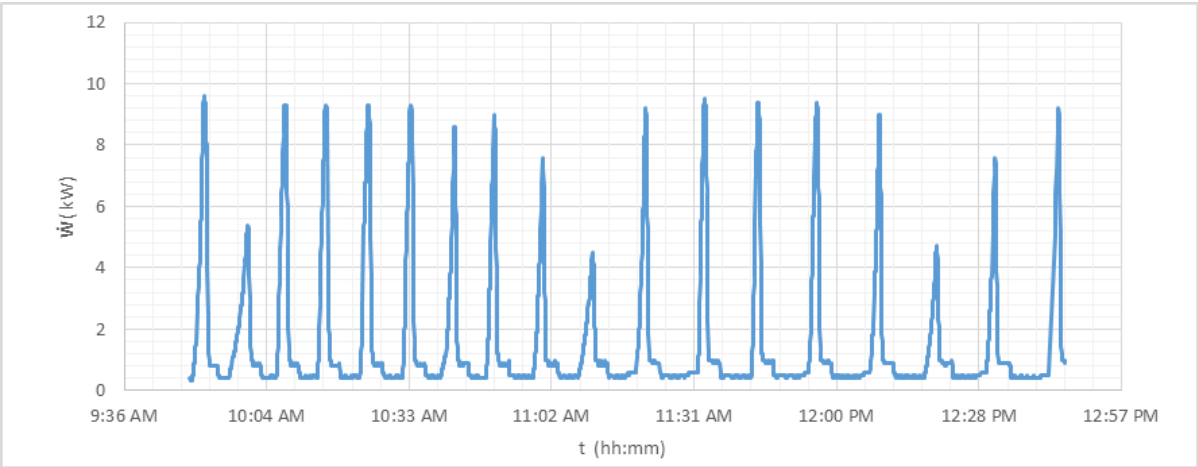


Fig. 61. Power consumption of Grundfos pumps when they ran from 9:30 am to 12:45 pm on February 23, 2015 for Ti of 0.5 second (reproduced from Ref. 1)

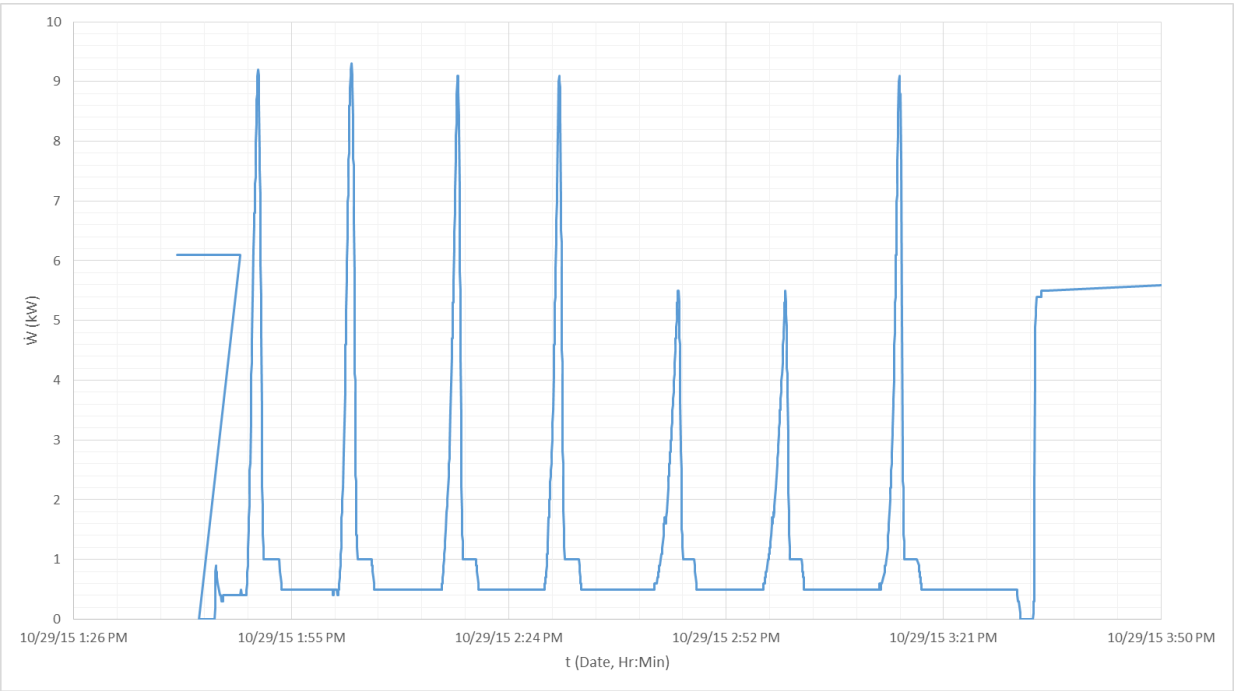


Fig. 62. Power consumption of Grundfos pumps when they ran from 1:30 to 3:30 pm on October 29, 2015 for Ti of 2 seconds

In the troughs between the peaks in Fig. 62, there was fairly constant power consumption which lasted for approximately 6-7 minutes. This was due to the fact that the pumps shifted from normal operation to minimal operation mode (i.e., 42% of full speed). The average power consumption for the Grundfos pumps was 3.46 kW for Fig. 62, again much less than the average power of 5.2 kW that the pumps consumed when they ran in pressure control mode from 12 pm to 1:30 pm (right before the level control mode was implemented).

As is evident from Fig. 63, the flow rate also peaked when the water level in the DA tank decreased below the 52% set point. The peaks were usually for very short time intervals ranging from 5 to 50 seconds. There was a very large flow of condensate into the DA tank during these peaks. When the last peak was examined, it was found that the average flow rate was 293.4 gpm, but for a time period of only 35 seconds. This caused the pumps to work much harder to maintain the water level in the DA tank at the 52% set point; and so the Grundfos pumps had higher power consumption during the peaks. The average flow rate for the Grundfos pumps when they ran in level control mode was 106.59 gpm (see Fig. 63).

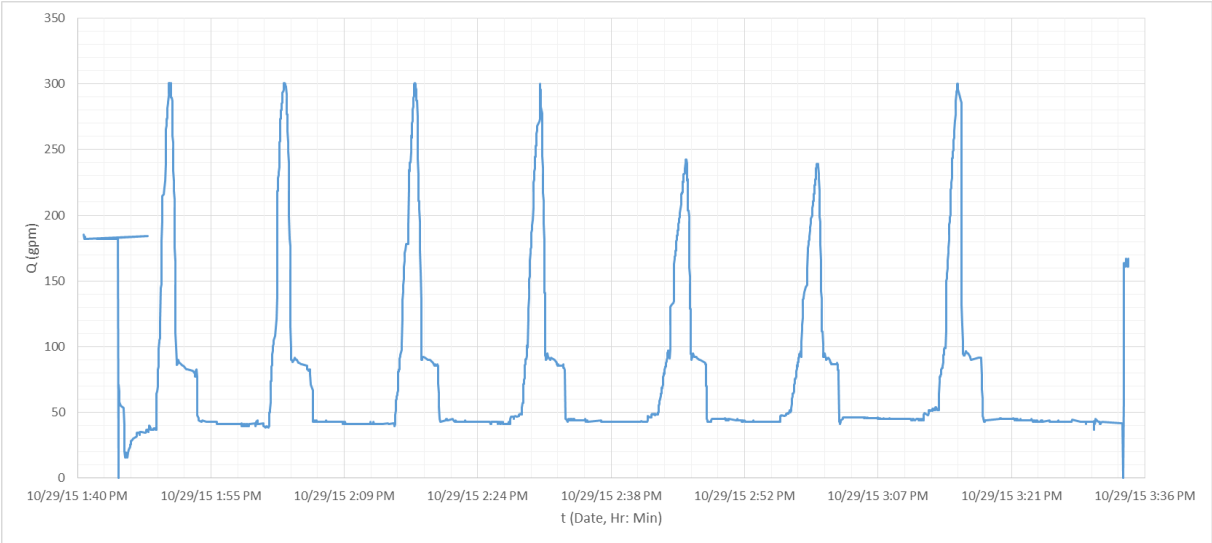


Fig. 63. Flow rate of condensate water when Grundfos pumps ran in level control mode on October 29, 2015

The initial peak in the Grundfos pumps' pressure, shown in Fig. 64, was because the control valve before the DA tank was not fully open. This forced the pumps to produce a high pressure in order to overcome the control valve's frictional head losses so that condensate water could reach the DA tank. The average discharge pressure was 14.16 psig for the Grundfos pumps when they ran in level control mode. The power consumed in the first and last peaks in Fig. 62 was excluded from the calculation of the average power consumption.

To summarize the data obtained for the pressure control (from 12 pm to 1:30 pm) and level control (from 1:30 pm to 3:30 pm) modes of the Grundfos pumps for October 21 and October 29 of 2015, Table 9 shows a comparison of the average discharge pressure, flow rate and the power consumption.

Because of the relatively colder weather on October 29 (mean temperature was 42 ^{0}F) as compared to that on October 21 (mean temperature was 55 ^{0}F), the steam demand was greater. Thus, the average power consumption and flow rates were higher for the Grundfos pumps on October 29, as compared to those of October 21. In level control mode, the Grundfos pumps consumed 76% less power on 21 October and 38% less power on October 29 as compared to the pressure control mode's power consumption values on the same dates.

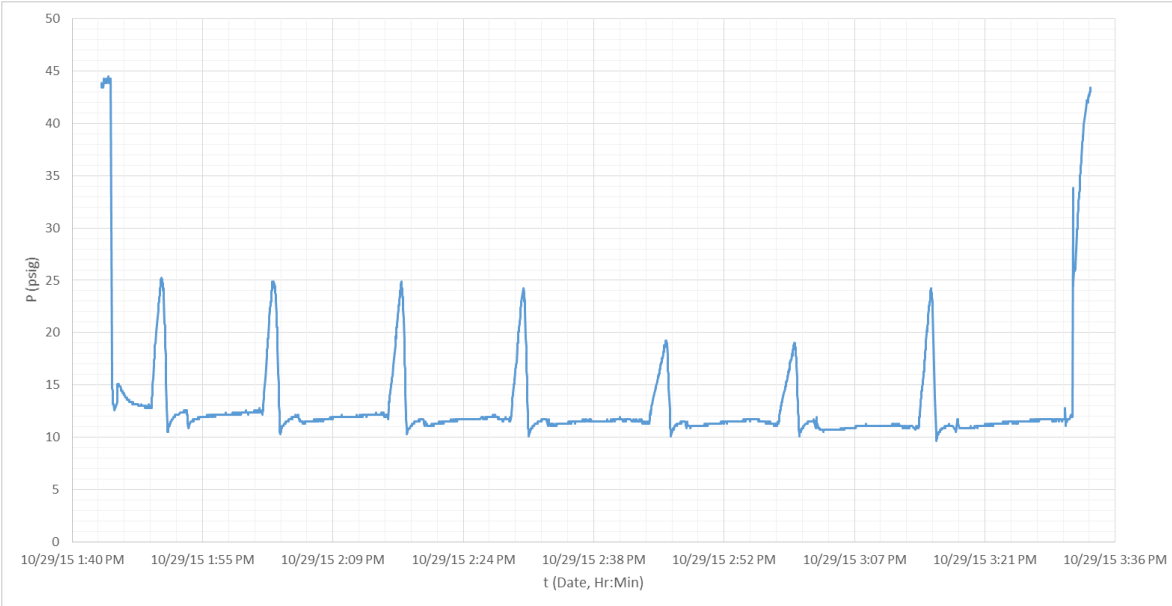


Fig. 64. Discharge pressure of Grundfos pumps when they ran in level control mode on October 29, 2015

Table 9. Comparison of data when Grundfos pumps ran in pressure control mode and level control mode on October 21 and October 29 of 2015

Grundfos Pump Mode	Avg. Discharge Pressure (psig)	Avg. Flow Rate (gpm)	Avg. Power Consumption (kW)
Pressure Control (October 21, 12 pm to 1:30 pm)	**42.71**	**142.34**	**5.25**
Level Control (October 21, 1:30 pm to 3:30 pm)	**12.94**	**64.08**	**1.33**
Pressure Control (October 29, 12 pm to 1:30 pm)	**42.83**	**141.56**	**5.21**
Level Control (October 29, 1:30 pm to 3:30 pm)	**14.16**	**106.59**	**3.46**

Just as was done for the Worthington pumps, the power used was validated by the same two methods, comparing the recorded power from the Grundfos control panel with: (1) power read from the Grundfos pump curve (see Fig. G2a), and (2) computed power using Eq. (9b). Table 10 shows a comparison of the power values along with the percentage differences.

For the Grundfos pumps, the power was recorded using the control panel data. In addition, power was calculated using Eq. (9b). Power was also read from the Grundfos pump performance curve (see Fig. G2a),

by inputting the flow rate (in gpm) and the discharge pressure (in ft. of head). Initially it was found that the Grundfos pumps had larger percentage errors (i.e., E and E') as compared to those of the Worthington pumps. This is because the Grundfos control panel recorded flow rate and power based on a Grundfos algorithm, instead of using the more accurate flow rate recorded by the Siemens flow meter. Since the Grundfos algorithm could not be trusted, Eq. (9b) used the flow rate recorded by the Siemens flow meter, instead of using the flow obtained from the algorithm.

When the flow rates recorded by the Siemens flow meter were used to determine power for both the Worthington and the Grundfos pumps, it was found that the Worthington pump actually had bigger errors as compared to those of the Grundfos pumps. There were negligible differences between power read from the pump curve and power calculated from Eq. (9b) (i.e., E'') for the Grundfos pump as compared to those for the Worthington pump. Equation (9a) was used instead of Eq. (9b) for the Worthington pump.

Aging could be a major factor in this analysis. The Grundfos pumps were installed in the power plant in 2010; however the Worthington pump was installed in 2005. Also, the efficiency from the Grundfos pump curve was the overall pump efficiency; but for the Worthington pump, it was unknown as to whether the pump curve showed the overall pump efficiency or was to be combined with the motor efficiency.

See Appendix H for the plots of the discharge flow rate and power consumption of both the Worthington and the Grundfos pumps, taken on the corresponding weeks from June of 2015 to November of 2015.

Table 10. Average power and percentage differences for power measurement of the Grundfos pumps

	June	July	August	September	October	November
Recorded Power from Control Panel (kW)	5.27	5.30	5.29	5.25	5.60	5.67
Power Read from Grundfos Pump Curve (kW)	4.39	4.54	4.54	4.49	4.76	4.79
Power Calculated from Eq. (9b) (kW)	4.41	4.53	4.53	4.47	4.73	4.77
E (%) (Difference between Recorded Power and Power read from Pump Curve)	-16.69	-14.34	-14.17	-14.47	-15.00	-15.52
E' (%) (Difference between Recorded Power and Calculated Power from Eq. (9b))	-16.31	-14.52	-14.36	-14.85	-15.53	-15.87
E'' (%) (Difference between Power Read from Pump Curve and Calculated Power from Eq. (9b))	-0.45	-0.22	-0.22	-0.44	-0.63	-0.41

4.iii Savings from Having Vent Condenser vs. Not Having Vent Condenser

This section deals with an assessment that was performed to see how much could be saved by using a less powerful pump (i.e., with no vent condenser present) as compared to the present system (i.e., with the Worthington condensate pump and the vent condenser present). The pump chosen was a Dayton 2ZWP2 centrifugal pump [22], which had a maximum discharge flow rate of 67 gpm at a pressure of 19 psig. For a flow rate of 50 gpm, the corresponding pressure was approximately 25 psig [22].

To make use of the vent condenser in the first floor, excess condensate water must be moved through an elevation change of 40 ft (i.e., 17.34 psig). So, the condensate pump should be powerful enough to provide water to both the vent condenser and the DA tank simultaneously. The Worthington constant speed pump had a discharge pressure of approximately 49 psig to accomplish both of these tasks.

However, if the vent condenser were not present at all or not used, then the condensate pump would have to supply water only to the DA tank. This would mean that the pump would not require a discharge pressure as high as 49 psig (see Table 4) to supply water to the DA tank, because the DA tank only required water at a pressure of approximately 10 psig (see Table 4). So, a less powerful pump (i.e., the Dayton pump) could achieve this task (i.e., supplying water only to the DA tank).

To calculate the work done to move water to the DA tank, use

$$\dot{W}' = C_2\, C_3 C_4\, Q\, \Delta P \,/\, \eta_p \tag{13}$$

C_2, C_3, and C_4 are conversion factors and are included in the Nomenclature. The average discharge flow rate for Worthington pump, over various months in 2015, was found to be approximately 140 gpm (refer to Table 5). Worthington pump efficiency was (see Fig. G1a), $\eta_p = 0.59$.

The average flow to the vent condenser, over various months in 2015, was approximately 90 gpm (refer to Table 5). Thus, the average flow rate to the DA tank was the difference between the flow rate of Worthington pump and the flow rate to the vent condenser or approximately ~50 gpm (refer to Table 5).

Using Eq. (13), the work required to move 140 gpm of water to the DA tank and the vent condenser (i.e., for the Worthington pump) was

$$\dot{W}_{DA,vent\ present}$$
$$= \left(0.133681 \frac{ft^3}{gallon}\right)\left(\frac{1}{778}\frac{Btu}{lb_f - ft}\right)\left(144\ \frac{in^2}{ft^2}\right)(140\ gpm)\ (49\ psig)\ \left(\frac{1}{0.59}\right)$$
$$= 287.67\ Btu/min$$

The work required to move 50 gpm of water to the DA tank (when no vent condenser is present, i.e., for the Dayton pump) would be

$$\dot{W}_{DA,vent\ absent}$$
$$= \left(0.133681 \frac{ft^3}{gallon}\right)\left(\frac{1}{778}\frac{Btu}{lb_f - ft}\right)\left(144\ \frac{in^2}{ft^2}\right)(50\ gpm)\ (25\ psig)\ \left(\frac{1}{0.59}\right)$$
$$= 52.41\ Btu/min$$

η_p for the Dayton pump was not available from its pump curve [22]. For the calculations, η_p was assumed to be the same as that for the Worthington pump, i.e., 0.59. Thus, the work required to move 90 gpm of water to the vent condenser is

$$\dot{W}_{net} = \dot{W}_{DA,vent\ present} - \dot{W}_{DA,vent\ absent}$$
$$= 287.67 - 52.41 = 235.26\ Btu/min$$

Energy reclaimed by the vent condenser was calculated by using a modified version of Eq. (2) where a conversion factor, C_2, is included.

$$E_{vent/HEX} = C_2\ m_{vent/HEX}\ c_p\ \Delta T_{rise} \tag{14}$$
$$= C_2\ \rho\ Q_{vent/HEX}\ c_p\ \Delta T_{rise}$$
$$= \left(0.133681 \frac{ft^3}{gallon}\right)\left(61.0\ \frac{lbm}{ft^3}\right)(90\ gpm)(1.001 \frac{Btu}{lb_m{}^{\circ}\mathrm{F}})\ (9^{\circ}\mathrm{F})$$
$$= 6611.78\ Btu/min$$

Excess condensate water moved to the vent condenser at a flow rate of 90 pm having an average temperature of 160^0 F and c_p = 1.001 Btu/lb_m ^{0}F. Using a modification of Eq. (4b), which includes an hour-minutes conversion factor, C_5, assuming a plant efficiency of 0.88 and the LHV_{fuel} of natural gas being 1018.6 Btu/ft^3, the volume of natural gas saved by using the vent condenser was

$$Vol_{fuel,saved} = \frac{C_5\ (E_{vent/HEX})}{(\eta_{plant})(LHV_{fuel})} \tag{15}$$
$$= \frac{(60\ \frac{min}{hr})\ (6611.78\ \frac{Btu}{min})}{(0.88)(1018.6\ \frac{Btu}{ft^3})} = 442.57\ \frac{ft^3}{hr}$$

Using the cost of natural gas as \$0.0053/ft^3 [17], the energy savings by using the vent condenser (i.e., using the Worthington pump) with respect to the baseline: the Worthington pump being used without the vent condenser, was

$$Hourly\ Vent\ Condenser\ Savings$$
$$= (Vol_{fuel,saved})\left({cost\ of\ natural\ gas}/{ft^3}\right) \tag{16}$$
$$= \left(442.57\ \frac{ft^3}{hr}\right)\left(\frac{\$0.0053}{ft^3}\right) = \$2.34/\mathrm{hr}$$

Using the cost of electricity as \$0.0717/kW-hr [17], if there were no vent condenser (i.e., using a Dayton pump), energy savings with respect to the baseline: the Worthington pump being used without the vent condenser, would be

$$\begin{aligned} &Hourly\ \ Savings\ (No\ Vent\ Condenser) \\ &= C_5\, C_6\, (\dot{W}_{net}) \left({cost\ of\ electricity}/{kW - hr} \right) \\ &= (60\, \tfrac{min}{hr})\ \left(0.000293\ \tfrac{kW-hr}{Btu}\right)(235.26\, Btu/\mathrm{min})\ (\$0.0717/\mathrm{kW-hr})\ =\ \$0.286/\mathrm{hr} \end{aligned} \tag{17}$$

Yearly vent condenser savings was calculated by multiplying \$2.34/hr by 8760 hr/yr. For one year, the savings by using the vent condenser was found to be \$20,498. If no vent condenser were present, the yearly savings was calculated by multiplying \$0.286/hr by 8760 hr/yr. For one year, the savings by not having the vent condenser was found to be \$2,505.

Because the two cases use different pumps, the effective annual costs of the pumps have to be included in the calculations in order to obtain the overall savings. The effective annual cost of the vent condenser also has to be taken into account.

To do this, the present value of the piece of equipment, the rate of interest and the number of compounding periods are required so as to be able to compute the effective uniform series value A_0. Effective uniform series value A_0 of any piece of equipment can be calculated by [23]

$$A_0 = \frac{PV\ d(1+d)^{ni}}{(1+d)^{ni} - 1} \tag{18}$$

This equation is used to consider the Worthington pump, the Grundfos pump and the vent condenser as the pieces of equipment. Here d is the rate of interest (assumed to be constant), ni is the number of compounding periods. In this case, since yearly compounding is assumed, ni is the expected life of each pump (20 years).

For the Worthington pump, *PV* is \$2500 [1]; so from Eq. (18), A_0 is \$168 for a 3% interest rate. For the Dayton pump, *PV* is \$524 [22]; so from Eq. (18), A_0 is \$35. For the vent condenser, PV is \$500 [4]; so from Eq. (18), A_0 is \$34 for a 3% interest rate.

Tables 11 and 12 show the savings from using the Worthington pump (i.e., vent condenser calculations are present) vs. the savings from using the Dayton pump (i.e., vent condenser not present). Both savings calculations are with respect to the baseline: the Worthington pump being used without the vent condenser.

For a rate of interest of 3%, the presence of the vent condenser yields \$20,296 in yearly savings as compared to just \$2,470 if there were no vent condenser present and the pump were smaller. For a rate of interest of 6%, the presence of the vent condenser yields \$20,236 in yearly savings as compared to just \$2,460 if there were no vent condenser present. All of the savings figures were calculated with respect to the baseline: the Worthington pump being used without the vent condenser. Thus, it can be concluded that the benefits of the vent condenser outweigh those of purchasing and maintaining a less powerful

pump. It can also be seen that the difference between interest rates does not cause much of a difference in the calculations. An important note is that maintenance costs are not included in the calculations. This is because they are assumed to be the same, regardless of the pump type and the vent condenser being present or absent. These tables assume that there is no change in the cost of electricity or gas for 20 years. Different results would be obtained if all of these changes are taken into account.

Table 11. Comparison of savings when vent condenser is present vs. not having the vent condenser for an assumed rate of interest of 3% over 20 years

	Worthington Pump	Dayton Pump
Annual energy savings from vent condenser ($)	20,498	0
Annual energy savings by using Dayton pump ($)	0	2,505
Yearly cost of pump ($)	-168	-35
Yearly cost of vent condenser ($)	-34	0
Total yearly savings ($)	**20,296**	**2,470**

Table 12. Comparison of savings when vent condenser is present vs. not having the vent condenser for an assumed rate of interest of 6% over 20 years

	Worthington Pump	Dayton Pump
Annual energy savings from vent condenser ($)	20,498	0
Annual energy savings by using Dayton pump ($)	0	2,505
Yearly cost of pump ($)	-218	-45
Yearly cost of vent condenser ($)	-44	0
Total yearly savings ($)	**20,236**	**2,460**

Potentially, the Grundfos pumps in level control mode could save more than the hypothetic Dayton pump because the Grundfos pumps only provide the pressure needed. However, the capital cost of the Grundfos pumps (i.e., approximately $2,640) is much higher than that of the Dayton pump (i.e., approximately $524). Ultimately, the purchase of pumps depends on the budget of the KU plant.

Chapter 5: Conclusions and Recommendations

5.i Conclusions

Both of the heat exchangers that were used in the power plant led to reduction in consumption of natural gas. This resulted in increased temperature of the boiler feed water. Due to this, there was an increase in plant efficiency from 0.86 to 0.88 in October, 2015 and from 0.92 to 0.94 in November, 2015 (see Table E1). The basement heat exchanger yielded an estimated savings of $520 for October of 2015 and $700 for November of 2015. Meanwhile, the vent condenser allowed for savings of $1,875 for October of 2015 and $2,530 for November of 2015. The yearly savings for the basement heat exchanger was estimated at $7,660 while, for the vent condenser, the estimate was $27,680.

Also, the monetary benefit of having a vent condenser with a powerful condensate pump far outweighed the benefit of having a less powerful condensate pump without a vent condenser. For a 3% interest rate, the yearly savings of the power plant when the vent condenser was present equaled $20,296, while that for the less powerful pump (Dayton pump) equaled $2,470; both savings were computed with respect to the baseline: the Worthington pump being used without the vent condenser. So, the power plant staff were justified in using the vent condenser.

The heat exchanger in the basement was a steam-water combination heat exchanger. Thus, this was not the ideal heat exchanger to use because there was water flow in both the shell and tube sides, instead of having steam flow in the shell side. It was also unknown whether either of the heat exchangers were parallel type flow or counter-flow. By assuming an efficiency of 0.9 and increasing the number of tubes to 4, the calculated area and the tube surface area values have a minor difference of 2.6%. So, it can be stated that the calculations validate the counter-flow assumption.

When running in level control mode, the Grundfos pumps' reaction to a water level change was not ideal. There were significant fluctuations in the power consumption and pressure of the Grundfos pumps due to large changes in flow rate. For certain time periods, the pumps consumed too much energy (i.e., as high as 8-10 kW), due to increased pump rotational speeds and high flow rates to the DA tank. Even so, the average power consumption when the Grundfos pumps ran in level control mode was 1.33 kW on October 21, 2015 and 3.66 kW on October 29, 2015. In comparison, the power consumption of the Grundfos pumps were 5.2 kW and 5.6 kW during pressure control mode on those two respective dates.

In conclusion, the Grundfos pumps' level control mode's power consumption was much less than that of the pumps' pressure control mode, even for colder weather. There were fewer fluctuations in the power consumption as compared to the results of Alabdullah [1]. This was because of changing the delay time Ti of the pumps from the default of 0.5 second to 2 seconds, which allowed the Grundfos pumps more time to respond to changes in the DA tank water level.

5.ii Recommendations for Future Work

- Presently, the power plant staff operate the mechanical valve at the outlet of the basement heat exchanger for the makeup water flow, based on visually evaluating the level of water in the condensate storage tank. An automated level sensor could be installed so that it detects the level of water in the storage tank, and an associated control valve could be installed in addition to the mechanical valve. This would eliminate the need for the solenoid valve. The mechanical valve could also be retained for backup emergency purposes.
- The steam power plant's boilers run at a pressure of 170-175 psig. However, the boiler feed pumps have discharge pressures as high as 350 psig. There is a huge potential for energy savings if the constant speed pumps can be replaced by variable speed pumps. A detailed study could be performed regarding the opportunity for these savings. For a flow rate of 50 gpm to the boiler and discharge pressure of 175 psig, the electricity saved is $0.46/hr, and so the yearly savings in electricity is approximately $4000 as compared to a pump whose discharge pressure is 350 psig.
- Instead of having the vent condenser on the first floor (where it is located presently), it should be moved to the basement at a location just above the DA tank. This would reduce the amount of extra pump pressure (i.e., 17 psig) required to lift the condensate water to the first floor. If the vent condenser were moved to the basement, the level control mode of the Grundfos pumps could be permanently used to provide condensate water to both the DA tank and the vent condenser. For an average flow rate of 64 gpm and average discharge pressure of 25 psig during level control mode of the Grundfos pumps with the vent condenser in the basement, approximately $15,000 can be saved in natural gas for 20 years as compared to that when the vent condenser is on the first floor.
- Temperature sensors that can be installed directly in the pipelines could be used. This could provide more accurate measurements of the condensate water temperatures which would make energy savings calculations in this thesis more accurate.
- The steam flow rate from the DA into the vent condenser is unknown. This steam condenses into water, which then flows to the storage tanks in the basement. A flow rate measuring device could be installed in this condensed water return line from the vent condenser to the storage tanks so as to record the condensed water's flow rate. There is also no flow rate data available for the hot water that flows from the flash tank to the basement heat exchanger. A flow rate measuring device could be installed in this pipe line. Knowledge of the flow rates of all fluids for the vent condenser and the basement heat exchanger would make the energy savings calculations more accurate.
- A flow rate measuring device with associated data acquisition system could be installed in the makeup water line. This would help in recording the makeup water flow rate, rather than estimating the makeup water flow rate based on the temperature of the water being used with hand-written data, thus improving accuracy in energy savings calculations.
- A reverse osmosis system could be installed in the power plant. This would help in reducing the amount of TDS in the water in the DA tank. This would also reduce the amount of boiler blowdown, because boiler blowdown is mainly done to remove the TDS in the boiler feed water.

- The water from the flash tank is drained to the sewers presently. This water has a significant amount of energy since it usually is at temperatures ranging from 100-140 ^{0}F. An extra heat exchanger could be installed so as to recover more energy from this water.
- The water returning from the campus could be run directly through the vent condenser on the first floor before it flows into the condensate storage tanks in the basement. This would eliminate the need for the condensate pumps to move excess condensate to the vent condenser. A less powerful pump could then be used to solely focus on providing water to the DA in the basement. However, to move the returning water from the campus to the vent condenser on the first floor, a pump that can supply at least 20 psig discharge pressure would be required.

Chapter 6: References

1. Alabdullah, R.J., Evaluation of Grundfos CRE 15-3 Variable Speed Centrifugal Pump and Worthington Constant Speed Centrifugal Pump Applications in KU Steam Power Plant, Department of Mechanical Engineering. May 2015 (Mech. Eng. M.S. Thesis); University of Kansas; Lawrence, KS.

2. Waid, V., Power Plant Supervisor. Personal Communication, June 2015; KU Steam Power Plant; Lawrence, KS.

3. O'Connor, C., Water Deaerating Apparatus. 1956; Google Patents; US2741327 A.

4. Mills, R., Power Plant Operator. Personal Communication, June 2015; KU Steam Power Plant; Lawrence, KS.

5. Babcock and Wilcox Company, Steam, Its Generation and Use. 2015; Babcock & Wilcox, Charlotte, NC.

6. Hokanson, A.E., Vent Condenser. 1984; Google Patents; US4471836 A.

7. Schmidt, F.P., Comparative Analysis between Grundfos CRE 15-3 Variable Speed Centrifugal Pumps and a Worthington D-824 Constant Speed Centrifugal Pump in a KU Steam Power Plant Application, Department of Mechanical Engineering. April 2014 (Mech. Eng. M.S. Thesis); University of Kansas; Lawrence, KS.

8. Tray Type Pressurized Deaerator - Operation. Available from: http://www.industrialsteam.com/content/products/pressurized-deaerators/tray-type-operation.asp; Date accessed: 11/27/2015.

9. Sherzer, A., Net Positive Suction Head. 1923; Google Patents; US1465097 A.

10. Nishio, M., J. Itoh, K. Shiroko and T. Umeda, A Thermodynamic Approach to Steam-Power System Design. Industrial & Engineering Chemistry Process Design and Development, 1980; **19**(2): pp. 306-312.

11. LLP, A.A.S. Deaerator & Feedtank Systems. Available from: http://in.ari-armaturen.com/fileadmin/user_upload/ari-in/files/Deaerator-brochure_01.pdf; Date accessed: 12/6/2015.

12. Kelly, T.J., Boiler Blowdown System. 1981; Google Patents; US4285302 A.

13. Sulzberger, K.J., Shell and Tube Heat Exchanger. 1989; Google Patents; US4871014 A.

14. Gleick, P.H., Water and Energy. Annual Review of Energy and the Environment, 1994; **19**(1): pp. 267-299.

15. Tallarida, R.J. and R.B. Murray, Area under a Curve: Trapezoidal and Simpson's Rules, in Manual of Pharmacologic Calculations. 1987; Springer. pp. 77-81.

16. El-Wakil, M.M., Powerplant Technology. 1988; Tata McGraw-Hill Education, New York City, NY.

17. Werth, G., Campus Energy Engineer, P.E. Personal Communication, October 2015; KU Facilities Service; Lawrence, KS.

18. Bergman, T.L., F.P. Incropera, and A.S. Lavine, Fundamentals of Heat and Mass Transfer. 2011; John Wiley & Sons, New York City, NY.

19. Bledsoe, D.M., Sales Engineer, HVAC. Email Communication, December 2015; Bell & Gosset; Kansas City, MO.

20. Karassik, I.J., J.P. Messina, P. Cooper and C.C. Heald, Pump Handbook. Vol. 3. 1986; McGraw-Hill, New York City, NY.

21. James, B., Business Development Manager. Personal Communication, September 2015; Grundfos Pumps Corporation; Olathe, KS.

22. Dayton Centrifugal Pump. Available from: https://www.grainger.com/product/DAYTON-Centrifugal-Pump-2ZWP2?functionCode=P2IDP2PCP; Date Accessed: 1/27/2015.

23. Fuller, S.K., Life-Cycle Costing Manual for the Federal Energy Management Program, S.R. Petersen, Editor. 1996, U.S. Dept. of Commerce, Technology Administration, National Institute of Standards and Technology: Gaithersburg, MD. Available from: http://fire.nist.gov/bfrlpubs/build96/PDF/b96121.pdf; Date Accessed: 12/12/2015.

24. Martin, Barry J., Boiler Control Systems. 1981; Google Patents; US4276857 A.

25. Operating Controls. Available from: http://www.nationalboard.org/Index.aspx?pageID=134; Date accessed: 5/7/2016.

26. TMC6-HE Temperature Sensor. Available from: http://www.onsetcomp.com/products/sensors/tmc6-he; Date accessed: 11/5/2015.

27. SITRANS FM MAG 5100 W. Available from: http://w3.siemens.com/mcms/sensor-systems/en/process-instrumentation/flow-measurement/electromagnetic/pulsed-dc-meters/sensors/pages/sitrans-f-m-mag-5100-w-for-water-applications.aspx; Date accessed: 11/5/2015.

28. ONSET HOBO Data Loggers. Available from: http://www.onsetcomp.com/products/data-loggers/u12-006; Date accessed: 11/25/2015.

29. Cadillac Meter. Available from: http://cadillacmeter.com/products/cadillac-cmag-magnetic-flow-meter/; Date accessed: 12/1/2015.

30. Danfoss Pressure Transducer. Available from: http://sensors.danfoss.com/pressure-transmitters/mbs3000/; Date accessed: 12/1/2015.

31. Omega Pressure Sensor. Available from: http://datasheets.globalspec.com/ds/2/OMEGAEngineering/FCB787E6-282F-488A-85B2-33A332260B77; Date accessed: 11/28/2015.

32. Grundfos Pressure Sensor. Available from: http://net.grundfos.com/Appl/ccmsservices/public/literature/filedata/Grundfosliterature-2667239.pdf; Date accessed: 11/22/2015.

33. Veris Power Monitor Sensor. Available from: http://www.veris.com/docs/Installs/h8041-h8044_i0f2.pdf; Date accessed: 1/16/2016.

34. Mastech Power Supply. Available from: http://www.mastechpowersupply.com/mastech-regulated-variable-dc-power-supply-hy3003d-30v-3a.html; Date accessed: 1/16/2016.

35. Suresite Level Transducer and Visual Indicator. Available from: http://www.gemssensors.com/Level/Visual-Indicators/SureSite/SureSite-Level-Indicators-Standard-Alloy; Date accessed: 11/27/2015.

36. ONSET HOBO Data Logger. Available from: http://www.onsetcomp.com/products/data-loggers/ux120-006m; Date accessed: 12/27/2015.

37. HYDRO MPC E 2CRE15-03 3X460V BASIS - 95055132. Available from: https://product-selection.grundfos.com/product-detail.html?from_suid=1453911274865003278074227273464&hits=1&productnumber=95055132&qcid=73841049&searchstring=95055132; Date accessed: 1/3/2016.

Appendix A: Boiler Control Systems

There are four boilers in the KU power plant (refer to Fig. 7a). These boilers are managed by control systems. Currently, there is a Plant Master (see Fig. A1) that receives input from the steam drum [4].

Most boilers of medium to high pressure today use “closed loop control systems” in their boiler control systems. A closed loop control system compares the process variable (PV) with the set point (SP). Based on the difference between the PV and the SP, a feedback signal is sent to the finite control elements (like control valves). These finite control elements then increase or decrease the PV so that it matches the SP [24].

The information described in this Appendix was obtained from the control systems manual for the power plant [25] and also from the power plant staff [4].

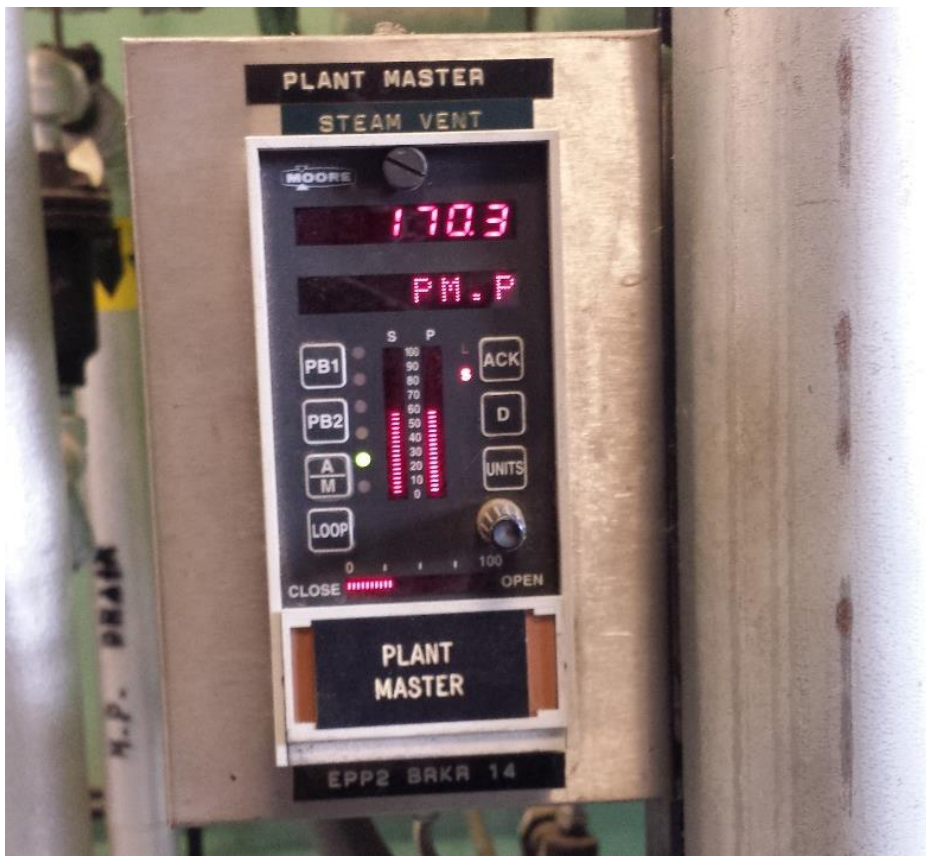


Fig. A1. Plant Master control system

The KU power plant uses various control systems like the Plant Master, Boiler Master, Gas Flow, Oil Flow, Oxygen Trim and Steam Drum Water Level loops. The Plant Master loop controls the Boiler Master loop, while the Boiler Master loop controls the Gas Flow and Oil Flow loops (see Fig. A2). The Oxygen Trim and the Steam Drum Water Level loops are independent [25]. The Process Variables associated with these loops are:

- Liquid level in the steam drum in inches
- Flow of feed water to the steam drum in gpm

- Flow of steam leaving the steam drum in kpph
- Pressure in the steam drum in psig

Proper operating control may be defined as "the ability to control a process variable at a given set point within an acceptable degree of accuracy" [25]. If not properly set up, abrupt changes in a set point can cause system controls to oscillate with excessive errors between SP and PV. Controllers are tuned so that the process variable is smooth and matches the set point. The information in the following two paragraphs is obtained from the power plant staff [4].

Fig. A2. Boiler control systems

Steam header pressure is the key variable that indicates the state of balance between the steam supply and the steam demand. If supply exceeds demand, the pressure will rise and vice versa. The term "Plant Master" is used when two or more boilers supply steam to a common steam header. Thus, there are multiple Boiler Masters but only one Plant Master. The Plant Master generates the master firing demand signal that drives the individual boilers [24]. The set point of the steam pressure in the steam header is 175 psig. According to Robert Mills, one of the power plant staff members [4], the Plant Master sends an output signal to the Boiler Master. The Boiler Master receives this signal. Then the Boiler Master sends output signals to the control systems for natural gas and oil. The oxygen trim system is independent and receives its input from the oxygen analyzer. The oxygen trim system offsets air flow to maintain optimum oxygen levels based on combustion calibration. These control systems (see Fig. A2) open/close their respective valves to allow/restrict their respective products to flow into the boiler.

According to Robert Mills [4], if the Plant Master receives an input signal that the steam pressure is 169 psig, it will essentially try to increase the fuel flow by opening the natural gas control valve more. Because of combustion of more fuel, there is an increase in temperature in the boiler. The steam pressure tries to reach 175 psig, which is the set point. For more steam to be generated, more fuel is required (i.e., natural gas). The Plant Master sends an output signal to the Boiler Master. After receiving an input signal from the Plant Master (in Fig. A2, the signal value is 37.0), the Boiler Master then transmits an output signal to the gas flow system (shown in Fig. A2). The gas flow control system receives the input signal and then

sends an output signal to its control valve to open more so that more natural gas flows into the boilers; thus steam generation is increased [4].

In the main office of the Power Plant, there is a computer display, where all of the respective control signals are shown (see Fig. A3). The drum level control system (which controls the feed water) is independent of all other systems. Here, the set point is 0.0, which means that the level of water in the steam drum is exactly at 50% full by volume. If the level of water becomes less than 50%, a negative .p value will be displayed which would then cause the control valve to open a little more so that more water can flow into the steam drum from the DA tank. The opposite occurs if the .p value exhibits a positive value.

Table A1 lists the various functions of the different push buttons/regulators for the Master control system, while Table A2 lists the meaning of the loop signals.

Table A1. Functions of the push buttons/regulators for the control system [4]

Numeric Display (6 digits)	Displays the numeric value of the Process Variable (PV) identified by the 8 character alphanumeric display. In Fig. A1, 170.3 is the steam pressure (in the steam drum) in psig.
Alphanumeric Display (8 characters)	Displays the loop name with associated process variable in the 6 digit numeric display. In Fig. A1, PM stands for Plant Master loop and .P stands for the Process Variable (PV) of that loop (which is the pressure in the steam drum).
PB1	Stands for proportional band. It is a range in percent from 0- 100%. When there are abrupt changes in PV, PB1 is changed so as to achieve stability in control during these system changes. For example, if the PV exceeds the SP, then the PB1 decreases accordingly so as to bring the PV back to its original SP.
PB2	It has the same function as PB1. It is only used when PB1 does not work or is faulty.
A/M	A stands for Automatic Control. M stands for Manual Control. Usually, the system works on A, but in case the operator wants to operate manually for a period of time for calibration or testing, M can be used.
LOOP	Pressing this button will advance to the next Active Loop, if more than one loop has been configured. For example, in Fig. A1, pushing the LOOP button has no effect because there is only one loop for this control system, i.e., the Plant Master loop.
ACK	Manages events within the controller. S- Indicates event in Station. L- Indicates event is active in Loop. Not much information is known about this parameter.
D	Changes the variable currently displayed, i.e., P, S, V, X values. (Explained in Table A2)
UNITS	Displays units of the variable shown in the alphanumeric display. For Fig. A1, it shows psig for the steam pressure.
S bargraph	Displays the scaled range of the controller SP for a specified variable as a percentage (in Fig. A1, the pressure in the steam drum).
P bargraph	Displays the scaled range of the controller PV as a percentage (in Fig. A1, the pressure in steam drum). In other words, it shows the operator whether the PV is more than or less than the controller SP. Ideally, the PV should be exactly the same as the SP.
V bargraph	Displays the scaled range of the Plant Master's controller output as a percentage of opening of the valve associated with the controller.

Table A2. Meaning of PM loop signals [4]

.P value	Stands for the Process Variable of the currently active loop.
.S value	Stands for the Set Point value of the currently active loop.
.V value	Stands for the Valve opening percentage of the currently active loop.
.X value	Stands for a optional variable value that is manually controlled by the user.

The different values of the data that the computer displays are shown in Fig. A3. For this snapshot, the gas (fuel) flow is 33.4 kpph (kilo-pounds per hour); the air flow is 37% of the total air available from the air compressors; the oxygen is 2.3% (by volume) of the air flow; the pressure in the steam drum is 177.3 psig; the level of water is 0.1 inch below the ideal water level in the steam drum; and the flow rates of the steam and the feed water are 31.3 kpph and 31.2 kpph, respectively.

Fig. A3. Boiler control system display on the main office computer of the power plant

Appendix B: Measuring Devices Used in the Project

Appendix B contains the brochures and/or technical specifications of the measuring instruments used for this thesis. The type of instrumentation used, the manufacturer and the model no. are listed in Table B1.

A Gateway laptop was used as part of the data acquisition system to record and plot the data obtained from the HOBO U12-006 and the HOBO UX120-006M data loggers. HOBOWARE software was installed in the Gateway laptop to analyze the data.

Table B1. The measuring instruments installed in the KU steam power plant by Schmidt [7], Alabdullah [1] and Nanda

Appendix No.	Type of Instrumentation	Manufacturer	Model No.
Appendix B1	Temperature sensor	ONSET	TMC6-HE
Appendix B2	Flow meter & transmitter	Siemens	SITRANS F M MAG 5100
Appendix B3	Data logger	HOBO	U12-006
Appendix B4	Flow meter	Cadillac	CMAG
Appendix B5	Pressure transducer	Danfoss	MBS 3000
Appendix B6	Pressure sensor	Omega	PX43E0-200GI
Appendix B7	Pressure sensor	Grundfos	DPI 0-2.5
Appendix B8	Power monitor sensor	VERIS	H804X
Appendix B9	Power supply	Mastech	HY3003D
Appendix B10	Level transducer & indicator	Suresite	Type BA
Appendix B11	Data logger	HOBO	UX120-006M
Appendix B12	Laptop	Gateway	LT2802u

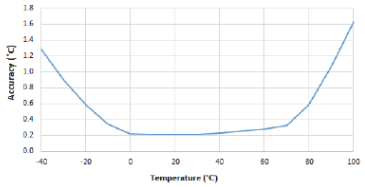
B1. TMC6-HE Temperature Sensor (reproduced from Ref. 26)

TMCx-HE Temperature Sensor

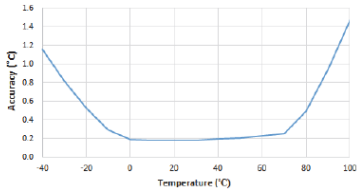
For use with HOBO® U12 and UX120-006M data loggers and ZW data nodes

Specifications

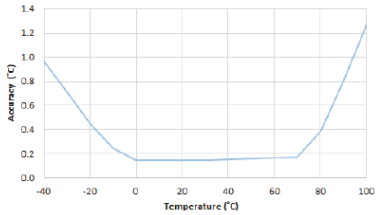
Measurement Range	-40° to 100°C (-40° to 212°F)
Accuracy with U12	±0.25°C from 0° to 50°C (±0.45°F from 32° to 122°F), insert probe 2.3 cm (0.9 inches) minimum; see Plot A
Accuracy with ZW	±0.21°C from 0° to 50°C (±0.38°F from 32° to 122°F), insert probe 2.3 cm (0.9 inches) minimum; see Plot B
Accuracy with UX120-006M	±0.15°C from 0° to 50°C (±0.27°F from 32° to 122°F), insert probe 2.3 cm (0.9 inches) minimum; see Plot C
Resolution with U12	0.03° at 20°C (0.05° at 68°F)
Resolution with ZW	0.02°C at 25°C (0.04°F at 77°F)
Resolution with UX120-006M	0.002° at 25°C (0.003° at 77°F)
Drift	<0.1°C (<0.2°F) per year
Response time in air	2 min. typical to 90% in air moving 1 m/sec (2.2 mph)
Response time on a pipe	Typically 2 times faster than the TMCX-HD. Typically less than 1 minute to 90%.
Housing	Copper-plated sensor tip
Dimensions	0.9 x 5.8 cm (0.38 x 2.30 inches)
Weight	34 g (1.1 oz)



Plot A: U12 Accuracy*



Plot B: ZW Accuracy*



Plot C: UX120-006M Accuracy*

*Accuracy shown in plots outside the 0 to 50°C range is typical.

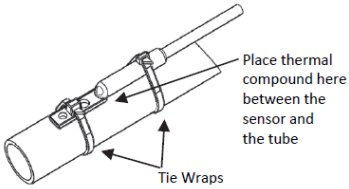
Items Included:

- 2 tie wraps
- 1 #6 screw
- Thermal compound

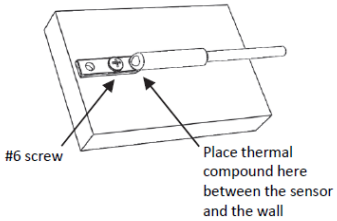
Mounting the Sensor

Use a small amount of the enclosed thermal compound between the flat part of the sensor tip and the surface where the sensor is being deployed to enhance the contact between the two.

Sensor mounted on pipe using two tie wraps:



Sensor mounted on wall using #6 screw:



B2. Siemens Flow Meter and Transmitter (reproduced from Ref. 27)

The SITRANS F M MAG 5100 W with its patented liners of hard rubber NBR or ebonite and EPDM is a sensor for all water applications such as ground water, drinking water, cooling water, waste water, sewage or sludge applications. Application examples: Water abstraction, Water distribution network, Waste water and as custody transfer water meter or cooling meter.

Details

Measuring range	0 to 10 m/s
Nominal Sizes	From DN 15 to DN 2000 (1" to 78")
Accuracy	0.2 % ±2.5 mm/s
Operating Pressure	Max. 16 bar (Max. 150 psi)
Ambient temperature	From -40 to 70 °C (-40 to 158 °F)
Medium Temperature	From -10 to 70 °C (14 to 158 °F)
Liners	EPDM NBR hard rubber Ebonite hard rubber
Electrodes	Hastelloy C-276 Built-in grounding electrodes
Material	Carbon steel, with corrosion resistant two-component epoxy coating
Drinking Water Approvals	EPDM: WRAS, NSF/ANSI Standard 61, DVGW 270, ACS and BelgAqua NBR: NSF/ANSI Standard 61, WRAS Ebonite: WRAS
Custody Transfer Approvals	OILM R 49 MI-001 PTB K7.2 (Germany) BEV OE12/C040 (Austria)
General approval	MCERTS Sira Certificate No. MC080136/00

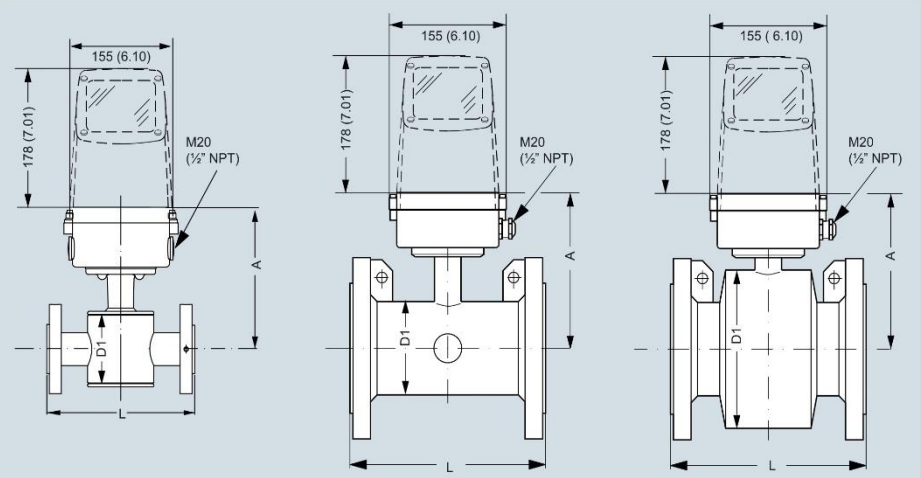
The SITRANS F M MAG 5000 is a microprocessor-based transmitter engineered for high performance, easy installation, commissioning and maintenance. The transmitter is truly robust, cost-effective and suitable for all-round applications and has a measuring accuracy of ± 0.4% of the flow rate (incl. sensor).

Application Examples: Water and waste water, General process industry, Food & beverage industry

Details

Accuracy	0.4 % ±1 mm/s
Input / output	1 current output 1 digital output 1 relay output
Communication	HART
Display	Background illumination with alphanumeric text, 3 x 20 characters
Enclosure	IP67 (NEMA 4x/6) IP20 (NEMA 2)
Power supply	12-24 V a.c./d.c. 115-230 V a.c.
Ambient temperature	From -20 to 50 °C (-4 to 122 °F)
Approvals	MI-001 Danak PTB OIML R49
Ex-approvals	FM/CSA Class 1, Div 2

Dimensional drawings



7ME6520: DN 15 ... 40 (½" ... 1½")

7ME6520: DN 50 ... 300 (2" ... 12")

7ME6520: DN 350 ... 1200 (14" ... 48")
7ME6580: DN 25 ... 2000 (1" ... 78")

		7ME6520 NBR or EPDM liner				7ME6580 Ebonite liner					
Nominal size		A		D1		A		D1		L	
[mm]	[inch]	[mm]	[inch]	[mm]	[inch]	[mm]	[inch]	[mm]	[inch]	[mm]	[inch]
15	½	177	7.0	77	3.0	-	-	-	-	200	7.9
25	1	187	7.4	96	3.8	187	7.4	104	4.09	200	7.9
40	1½	202	8.0	127	5.0	197	7.8	124	4.88	200	7.9
50	2	188	7.4	76	3.0	205	8.1	139	5.47	200	7.9
65	2½	194	7.6	89	3.5	212	8.3	154	6.06	200	7.9
80	3	200	7.9	102	4.0	222	8.7	174	6.85	200	7.9
100	4	207	8.1	114	4.5	242	9.5	214	8.43	250	9.8
125	5	217	8.5	140	5.5	255	10.0	239	9.41	250	9.8
150	6	232	9.1	168	6.6	276	10.9	282	11.1	300	11.8
200	8	257	10.1	219	8.6	304	12.0	338	13.31	350	13.8
250	10	284	11.2	273	10.8	332	13.1	393	15.47	450	17.7
300	12	310	12.2	324	12.8	357	14.1	444	17.48	500	19.7
350	14	382	15.0	451	17.8	362	14.3	451	17.76	550	21.7
400	16	407	16.0	502	19.8	387	15.2	502	19.76	600	23.6
450	18	438	17.2	563	22.2	418	16.5	563	22.16	600	23.6
500	20	463	18.2	614	24.2	443	17.4	614	24.17	600	23.6
600	24	514	20.2	715	28.2	494	19.4	715	28.15	600	23.6
700	28	564	22.2	816	32.1	544	21.4	816	32.13	700	27.6
750	30	591	23.3	869	34.2	571	22.5	869	34.21	750	29.5
800	32	616	24.3	927	36.5	606	23.9	927	36.5	800	31.5
900	36	663	26.1	1032	40.6	653	25.7	1032	40.63	900	35.4
1000	40	714	28.1	1136	44.7	704	27.7	1136	44.72	1000	39.4
	42	714	28.1	1136	44.7	704	27.7	1136	44.72	1000	39.4
	44	765	30.1	1238	48.7	755	29.7	1238	48.74	1100	43.3
1200	48	820	32.3	1348	53.1	810	31.9	1348	53.07	1200	47.2
1400	54	-	-	-	-	925	36.4	1574	65.94	1400	55.1
1500	60	-	-	-	-	972	38.2	1672	65.83	1500	59.1
1600	66	-	-	-	-	1025	40.4	1774	75.39	1600	63
1800	72	-	-	-	-	1123	44.2	1974	77.72	1800	70.9
2000	78	-	-	-	-	1223	48.1	2174	85.59	2000	78.7

- not available

B3. HOBO U12-006 Data Logger (reproduced from Ref. 28)

onset
HOBO® Data Loggers

HOBO® U12 Logger

Multi-channel energy & environmental monitoring

HOBO U12 data loggers provide flexibility for monitoring up to 4 channels of energy and environmental data with a single, compact logger. They provide 12-bit resolution measurements for detecting greater variability in recorded data, direct USB connectivity for convenient, fast data offload, and a 43K measurement capacity.

Supported Measurements: Temperature, Relative Humidity, Dew Point, 4-20mA, AC Current, AC Voltage, Air Velocity, Carbon Dioxide, Compressed Air Flow, DC Current, DC Voltage, Gauge Pressure, Kilowatts, Light Intensity, Volatile Organic Compound (some sensors sold separately)

Key Advantages:

- Records up to 4 channels
- Your choice of three models, with flexible measurement options
- Programmable as well as push-button start
- Compatible with a broad range of external sensors

Minimum System Requirements:

Software USB cable*

For complete information and accessories, please visit: **www.onsetcomp.com**

Part number	U12-006 (4 Ext)	U12-012 (Temp/RH/Light/Ext)	U12-013 (Temp/RH/2 Ext)
Memory	43,000 measurements		
Sampling rate	1 second to 18 hours, user-selectable		
Battery life	1 year typical, user-replaceable, CR2032		
		Temperature	
Max range		-20° to 70°C (-4° to 158°F)	
Accuracy		± 0.35°C from 0° to 50°C (± 0.63°F from 32° to 122°F)	
Resolution (12-bit)		0.03°C @ 25°C (0.05°F @ 77°F)	
		Relative Humidity	
Measurement range		5% to 95% RH (non-condensing)	
Accuracy		± 2.5% typical, 3.5% maximum, from 10 to 90% RH	
Resolution (10-bit)		0.03% RH	
		Light Intensity	
		Designed for general purpose indoor measurement of relative light levels	
Range		1 to 3000 footcandles (lumens/ft2) typical 0-32,300 lumens/m2	
	External Input		
Range	0 to 2.5 VDC		
Accuracy	± 2 mV, ± 2.5% of absolute reading		
Resolution	0.6 mV		
CE compliant	Yes		

*USB cable included with software
For stand-alone data logging applications in harsh indoor environments, see the 4-channel HOBO U12 Industrial data logger (U12-008) at onsetcomp.com

Contact Us

Sales (8am to 5pm ET, Monday through Friday)
- Email sales@onsetcomp.com
- Call 1-800-564-4377
- Fax 508-759-9100

Technical Support
- (8am to 8pm ET, Monday through Friday)
- Contact Product Support
- Call 877-564-4377

B4. Cadillac Magnetic Flow Meter (reproduced from Ref. 29)

Cadillac Meter

ACCURATE & RELIABLE ENERGY METERS

GENERAL INFORMATION

Cadillac® Magnetic Flow Meter CMAG Series

CENTRAL STATION STEAM CO.® 15615 SW 74TH AVE., STE #150 TIGARD, OR 97224 PHONE: 888-556-3913 FAX: 503-624-6131 @ WWW.CADILLACMETER.COM

Rev 0613

METER INSTALLATION

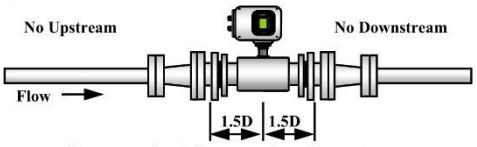
Cadillac CMAG Piping Requirements

Installation requirements have been redefined with the Cadillac magnetic flow meter.

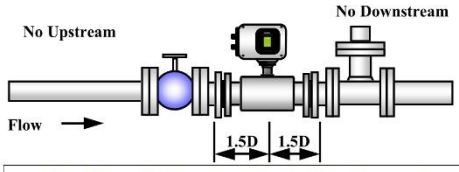
Employing coil and plate shaping techniques the Cadillac® meter provides a uniform magnetic flux shaping within the flow tube. This allows the meter to measure and sample uniformly the entire flow tube area.

In addition, the electronics provide high frequency DC square wave signal generation and flow signal sampling. When combined these two techniques eliminate all straight run flow profiling requirements, significantly decrease signal to noise ratio, and increase low flow accuracy.

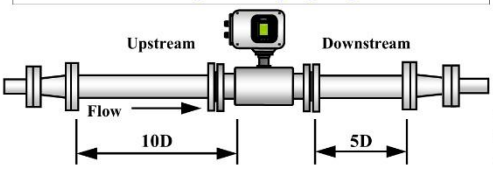
In practice, this means the Cadillac magnetic flow meters may be installed next to elbows, tees, valves, etc. without any effect in meter accuracy or stability. (See Illustrations). This also allows the CMAG to be installed in gravity flow applications with a turndown of 300:1 @ +/- 0.25% accuracy.



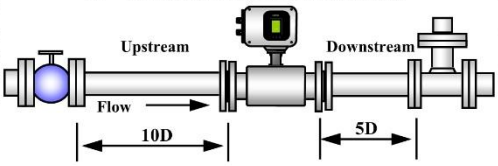
D = nominal flow meter diameter



Traditional Magmeter Piping Requirements



D = nominal flow meter diameter



In comparison, the straight pipe run requirements for all other magnetic flow meters are as follows:

Downstream of the meter:

- ◊ Expander (2-5) diameters
- ◊ Tee - (2-5) diameters
- ◊ Elbow - (2-5) diameters
- ◊ Valves - (2-10) diameters

- **Upstream of the meter:**
  - ◊ Expander - (10) diameters
  - ◊ Tee - (5) diameters
  - ◊ Elbow - (5) diameters
  - ◊ Valves - (10) diameters

Unlike other technologies such as the Cadillac® Vortex flow meter, magnetic flow meters do not have a low flow cutoff, essentially allowing the meter to read to zero. With such a wide flow range capability for the technology, most applications can be addressed with meters at full line size.

The Cadillac® magnetic flow meter has a 304 stainless steel body and is always sold with integral grounding rings installed. The primary reason for providing grounding rings is to contain the magnetic field within the meter body and to assure the liquid potential is grounded properly. As a consequence, the induced voltage is remarkably free of noise allowing the meter to reliably measure extremely low fluid velocities. The table below lists minimum and maximum 4-20 madc output spans for each meter size, in GPM for liquids and lbs/hr for condensate.

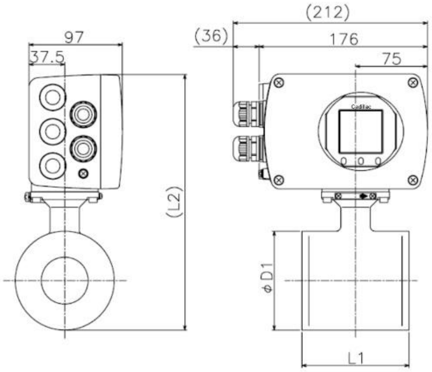
CMAG	Liquid Flow Range Table		Condensate Flow Range Table	
Meter Body (inches) Size	Minimum Volumetric (gal/min) Range	Maximum Volumetric (gal/min) Range	Minimum Condensate (lbs/hr) Range	Maximum Condensate (lbs/hr) Range
0.50"	0.00 - 0.250	0.00 - 25.00	0.00 - 125.0	0.00 - 12,500
1.0"	0.00 - 0.750	0.00 - 75.00	0.00 - 375.0	0.00 - 37,500
1.5"	0.00 - 1.750	0.00 - 175.0	0.00 - 875.0	0.00 - 875,000
2.0"	0.00 - 3.000	0.00 - 300.0	0.00 - 1,500	0.00 - 150,000
3.0"	0.00 - 8.000	0.00 - 800.0	0.00 - 4,000	0.00 - 400,000
4.0"	0.00 - 12.50	0.00 - 1,250	0.00 - 6,250	0.00 - 625,000
6.0"	0.00 - 25.00	0.00 - 2,500	0.00 - 12,500	0.00 - 1,250,000
8.0"	0.00 - 50.00	0.00 - 5,000	0.00 - 25,000	0.00 - 2,500,000
10.0"	0.00 - 75.00	0.00 - 7,500	0.00 - 37,500	0.00 - 3,750,000

Low velocity "Turndown Accuracy" of the CMAG has allowed it to address applications, which were not possible for volumetric flow meters in the past. Below is the turndown accuracy for the CMAG:

- ♦ **(+/- 0.25%)** of rate at **300:1** turndown with 1.5 diameters of straight piping from meter centerline up/downstream.
- ♦ **(+/- 0.50%)** of rate from **300:1 to 400:1** turndown with 1.5 diameters of straight piping from meter centerline up/downstream.
- ♦ **(+/- 1.00%)** of rate from **400:1 to 500:1** turndown with 1.5 diameters of straight piping from meter centerline up/downstream.

CENTRAL STATION STEAM CO.® 15615 SW 74TH AVE., STE #150 TIGARD, OR 97224 PHONE: 888-556-3913 FAX: 503-624-6131 @ WWW.CADILLACMETER.COM

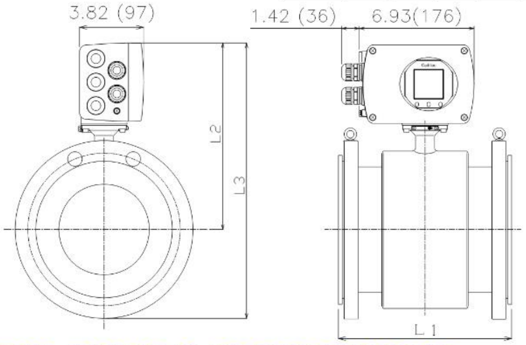
Rev 0613



Cadillac Meter
ACCURATE & RELIABLE ENERGY METERS

CMAG Flanged ANSI class 150 dimensions					
Meter Size (inch)	L1 (inch)	L2 (inch)	L3 (inch)	No. of Bolts	Weight (lbs)
1/2	5.51	7.99	9.88	4	21
1	6.30	8.23	10.71	4	25
1 1/2	6.69	8.58	11.34	4	28
2	7.09	9.13	12.20	4	32
3	9.06	10.04	13.66	4	62
4	9.45	10.31	14.45	8	72
6	10.24	11.22	16.73	8	115
8	11.81	12.20	18.70	8	145
10	13.78	13.18	21.06	12	210
12	15.75	14.37	23.15	12	355
14	17.72	14.72	24.37	12	460

Unit: inch (mm)



CMAG Wafer ANSI class 150/300 dimensions				
Meter Size (inch)	L1 (inch)	L2 (inch)	D1 (inch)	Weight (lbs)
1/2	2.76	9.33	1.93	17
1	3.15	8.90	2.60	21
1 1/2	3.94	9.80	3.35	25
2	4.33	10.43	4.02	29
3	4.33	11.46	5.00	35
4	4.72	12.72	6.26	40

CADILLAC® MAGNETIC FLOW METER GENERAL SPECIFICATIONS

- Meter will consist of a full-bore body with encapsulated and rigidly retained set of coils.
- Meter available with remote or integral electronics with indication and totalization.
- Meter operates at ± 0.25% accuracy without flow profiling or piping straight run exceeding 1.5 diameters.
- Meter operates with minimum 300 to 1 turndown at stated operating accuracy.
- Meter available with pulse and analog (4-20 mA) outputs.
- Meter will provide instantaneous and totalized flow available at local indicator or remotely through outputs.
- Meter measures flow using Faraday's law (Induce voltage is directly proportional to the velocity of the conductive liquid)
- Meter K-factor is stable and not influenced by external piping or mounting orientation.
- Meter will have uniform magnetic field flux distribution eliminating piping straight run and flow profiling.
- Meter will measure fluids with conductivity greater than or equal to 3.0 uS/cm.
- Meter will be calibrated to/and provided with NIST calibration certificate.

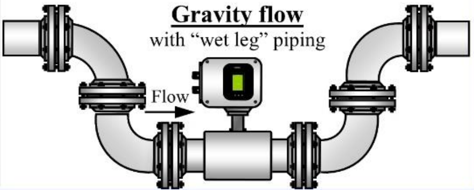
CADILLAC® MAGMETER MODEL NUMBER STRUCTURE

CMAG	Cadillac Magnetic Flow Meter
A	Size 0.5"
B	Size 1"
C	Size 1.5"
D	Size 2"
E	Size 3"
F	Size 4"
G	Size 6"
H	Size 8"
I	Size 10"
J	Size 12"
K	Size 14" (Consult factory for larger sizes)
II	Integral Converter with Indicator/Totalizer
RC	Remote Converter
W	Wafer Body (Ceramic Liner only) - available thru 4"
F	Flanged Body
150	ANSI Class 150
300	ANSI Class 300
S	Electrode Material - 316L SS (Std w/ Poly & EPDM)
H	Electrode Material - Hastelloy C (std w/PFA)
X	Electrode Material - Other
C	Liner Material - PFA (Teflon) - (Temp rated 250˚F)
U	Liner Material - Polyurethane (NSF approved)
D	Liner Material - EPDM (Rubber)
A	Liner Material - Ceramic (Temp rated 350˚F w/RC)
S	Ground Ring Material - 316 Stainless Steel
H	Ground Ring Material - Hastelloy C
X	Ground Ring Material - Other
FM	FM Approvals

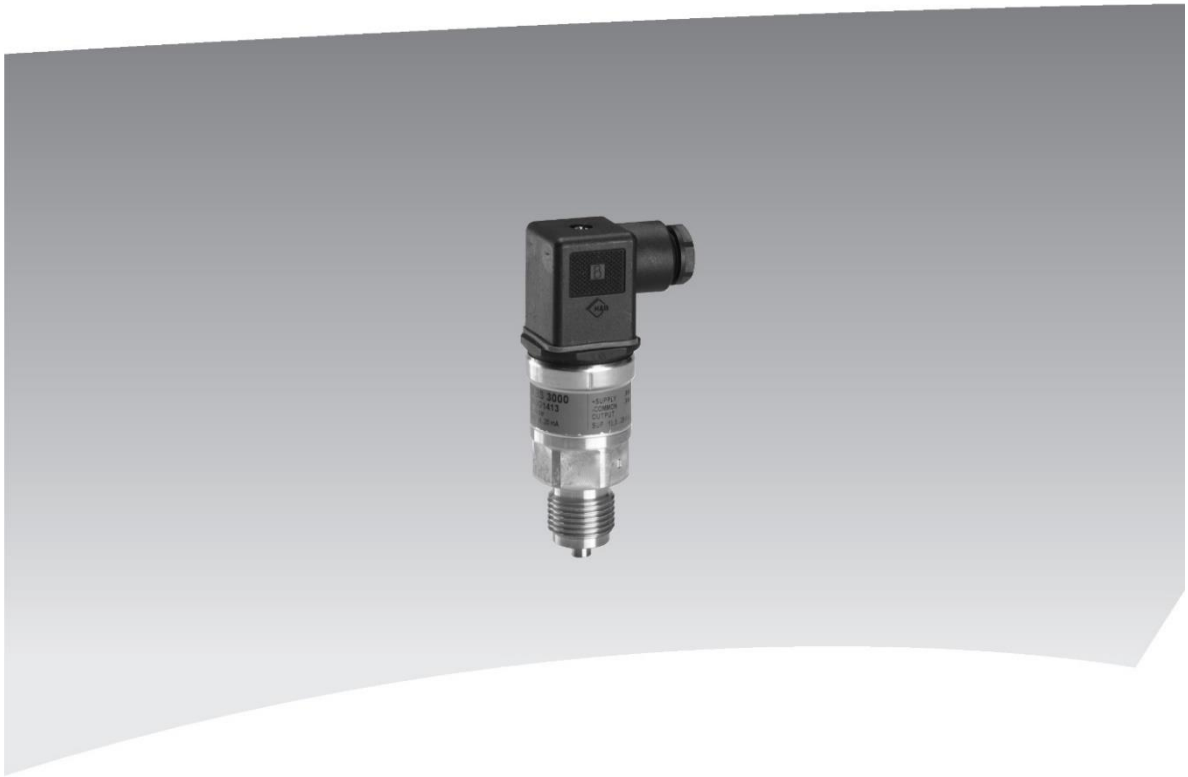
CMC	CMAG Remote Converter
R	Remote Mounting
I	Indicator/Totalizer
U	Universal Mounting Bracket
XXFT	Interconnecting Cable (length in feet)
FM	FM Approvals

APPLICATION NOTE:

The Cadillac® magnetic flow meter is an excellent choice for gravity flow applications. However for proper flow meter operation the piping and flow tube must be full at all times. This requires the meter to be mounted in such a way to provide a "Wet Leg" as illustrated below.



B5. Danfoss Pressure Transducer (reproduced from Ref. 30)



Pressure transmitter for industrial applications

Type MBS 3000

Technical brochure

Technical data

Performance (EN 60770)

Accuracy (incl. non-linearity, hysteresis and repeatability)	0.5% FS (typ.) 1% FS (max.)
Non-linearity BFSL (conformity)	≤ 0.2% FS
Hysteresis and repeatability	≤ 0.1% FS
Thermal zero point shift	≤ 0.1% FS/10K (typ.) ≤ 0.2% FS/10K (max.)
Thermal sensitivity (span) shift	≤ 0.1% FS/10K (typ.) ≤ 0.2% FS/10K (max.)
Response time	< 4 ms
Overload pressure (static)	6 × FS (max. 1500 bar)
Burst pressure	> 6 × FS (max. 2000 bar)
Durability, P: 10-90% FS	>10×10^6 cycles

Electrical specifications

	Nom. output signal (short-circuit protected)		
	4 – 20 mA	0 - 5, 1 - 5, 1 - 6 V	0 - 10 V , 1 - 10 V
Supply voltage [U_B], polarity protected	9 → 32 V	9 → 30 V	15 → 30 V
Supply - current consumption	–	≤ 5 mA	≤ 8 mA
Supply voltage dependency	≤ 0.05% FS/10 V		
Current limitation	28 mA (typ.)	–	
Output impedance	–	≤ 25Ω	
Load [R_L] (load connected to 0V)	$R_L \leq (U_B-9V)/0.02$ A	$R_L \geq 10$ kΩ	$R_L \geq 15$ kΩ

Environmental conditions

Media temperature range			–40 → +85°C
Ambient temperature range (depending on electrical connection)			see page 5
Compensated temperature range			0 → +80°C
Transport temperature range			–50 → +85°C
EMC - Emission			EN 61000-6-3
EMC Immunity			EN 61000-6-2
Insulation resistance			> 100 MΩ at 100 V
Mains frequency test			SEN 361503
Vibration stability	Sinusoidal	15.9 mm-pp, 5 Hz-25 Hz	IEC 60068-2-6
		20 g, 25 Hz - 2 kHz	
		Random 7.5 g_{rms},5 Hz - 1 kHz	IEC 60068-2-64
Shock resistance	Shock	500 g / 1 ms	IEC 60068 - 2 - 27
	Free fall		IEC 60068 - 2 - 32
Enclosure (depending on electrical connection)			see page 5

Mechanical characteristics

Materials	Wetted parts	EN 10088-1; 1.4404 (AISI 316 L)
	Enclosure	EN 10088-1; 1.4404 (AISI 316 L)
	Electrical connections	see page 5
Weight (depending on pressure connection and electrical connection)		0.2 - 0.3 kg

Ordering of special versions

MBS 3000 - _ _ _ _ _ _ _ _ _ _ _ _ _ _ _ _

Measuring range

0 - 1 bar	1 0
0 - 1.6 bar	1 2
0 - 2.5 bar	1 4
0 - 4 bar	1 6
0 - 6 bar	1 8
0 - 10 bar	2 0
0 - 16 bar	2 2
0 - 25 bar	2 4
0 - 40 bar	2 6
0 - 60 bar	2 8
0 - 100 bar	3 0
0 - 160 bar	3 2
0 - 250 bar	3 4
0 - 400 bar	3 6
0 - 600 bar	3 8

Pressure reference

Gauge (relative)	1
Absolute	2

Pressure connection

A B 0 4	G ¼ A (EN 837)
A B 0 6	G $^3/_8$ A (EN 837)
A B 0 8	G ½ A (EN 837)
A C 0 4	¼ -18 NPT
A C 0 8	½ -14 NPT
G B 0 4	DIN 3852-E-G ¼ Gasket: DIN 3869-14 NBR

Electrical connection

Figures refer to plug and standard PIN configuration - see page 5

1	Plug Pg 9 (EN 175301-803-A)
2	*)Plug, AMP Econoseal, J series, male, excl. female plug
3	Screened cable, 2 m
5	*)Plug, IEC 947-5-2, M12 × 1, male, excl. female plug
8	*) Plug, AMP Superseal 1.5 series male, excl. female plug

Output signal

1	4 - 20 mA
2	0 - 5 V
3	1 - 5 V
4	1 - 6 V
5	0 - 10 V
7	1 - 10 V

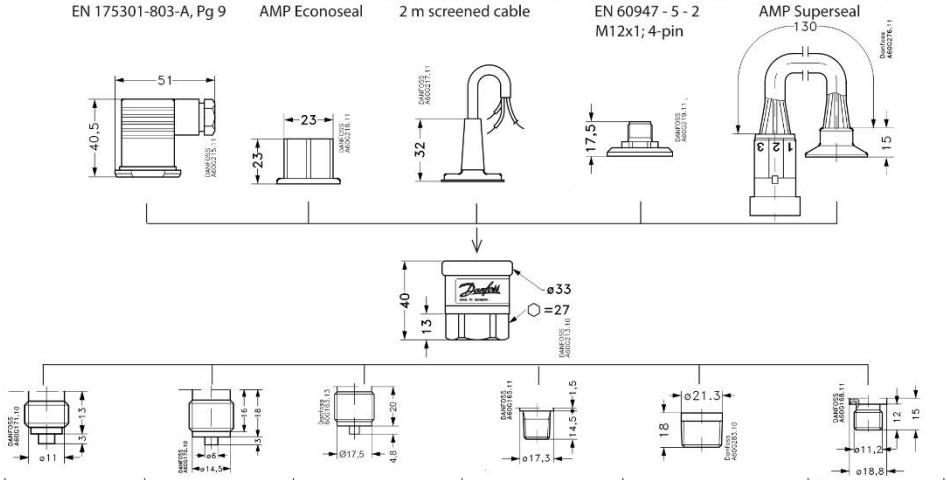
*) Gauge versions only available as sealed gauge versions

Preferred versions

Non-standard build-up combinations may be selected. However, minimum order quantities may apply. Please contact your local Danfoss office for further information or request for other versions.

Dimensions / Combinations

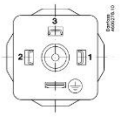
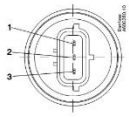
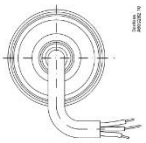
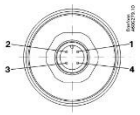
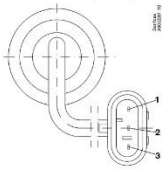
Type code	1	2	3	5	8
	EN 175301-803-A, Pg 9	AMP Econoseal	2 m screened cable	EN 60947 - 5 - 2 M12x1; 4-pin	AMP Superseal



	G¼ A (EN 837)	G $^3/_8$ A (EN 837)	G ½ A (EN 837)	¼ - 18 NPT	½ - 14 NPT	DIN 3852-E-G ¼ Gasket: DIN 3869-14
Type code	AB04	AB06	AB08	AC04	AC08	GB04
Recommended torque 1)	30-35 Nm	30-35 Nm	30-35 Nm	2-3 turns after finger tightened	2-3 turns after finger tightened	30-35 Nm

1)Depends of different parameters as packing material, mating material, thread lubrication and pressure level.

Electrical connections

Type code, page 4				
1	2	3	5	8
EN 175301-803-A, Pg 9 	AMP Econoseal J series (male) 	2 m screened cable 	EN 60497-5-2 M12x1 4-pin 	AMP Superseal 1.5 series (male) 
Ambient temperature				
−40 → + 85 °C	−40 → + 85 °C	−30 → +85 °C	−25 → +85 °C	−40 → +85 °C
Enclosure (IP protection fulfilled together with mating connector)				
IP 65	IP 67	IP 67	IP 67	IP 67
Materials				
Glass filled polyamid, PA 6.6	Glass filled polyamid, PA 6.6[1]	Poliolyfin cable with PE shrinkage tubing	Nickel plated brass, CuZn/Ni	Glass filled polyamid, PA 6.6[2]
Electrical connection, 4 - 20 mA output (2 wire)				
Pin1: + supply Pin 2: ÷ supply Pin 3: Not used Earth: Connected to MBS enclosure	Pin 1: + supply Pin 2: ÷ supply Pin 3: not used	Brown wire: + supply Black wire: ÷ supply Red wire: Not used Orange: Not used Screen: Not connected to MBS enclosure	Pin 1: + supply Pin 2: Not used Pin 3: Not used Pin 4: ÷ supply	Pin 1: + supply Pin 2: ÷ supply Pin 3 Not used
Electrical connection, 0 - 5V, 1 - 5 V, 1 - 6 V, 0 - 10 V, 1 - 10 V output				
Pin 1: + supply Pin 2: ÷ supply Pin 3: Output Earth: Connected to MBS enclosure	Pin 1: + supply Pin 2: ÷ supply Pin 3: Output	Brown wire: Output Black wire: ÷ supply Red wire: + supply Orange: Not used Screen: Not connected to MBS enclosure	Pin 1: + supply Pin 2: Not used Pin 3: Output Pin 4: ÷ supply	Pin 1: + supply Pin 2: ÷ supply Pin 3: Output

[1] Female plug: Glass filled polyester, PBT
[2] Wire: PETFE (teflon)
Protection sleeve: PBT mesh (polyester)

B6. Omega Pressure Sensor (PX43E0-200GI) (reproduced from Ref. 31)

HEAVY-DUTY FLUSH DIAPHRAGM TRANSMITTER
½ NPT THREAD

4 to 20 mA Output
0-50 to 0-750 psi
0-3 to 0-50 bar

1 bar = 14.5 psi
1 kg/cm² = 14.22 psi
1 atmosphere = 14.7 psi = 29.93 inHg = 760.2 mmHg = 1.014 bar

PX43E0-100GI, shown actual size.

CURRENT OUTPUT PRESSURE TRANSDUCERS B

PX43 Series

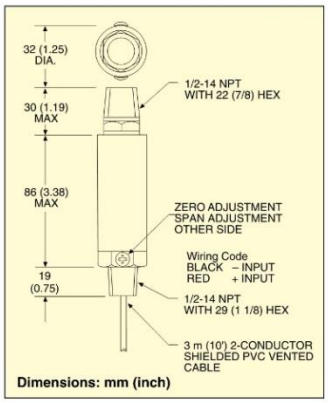
MADE IN USA NIST CE OMEGADYNE RoHS
Standard

- Stainless Steel Construction
- Processing or Industrial Applications
- Rugged Flush Diaphragm for Measurement of Difficult Fluids
- 3 m (10') Cable with Conduit Connection for Installation in Harsh Environments
- Heavy-Duty ½ NPT Fitting
- 4 to 20 mA Output for Noise-Free Transmission

OMEGA's PX43 is a high-accuracy, current output, industrial pressure transmitter with a heavy-duty flush diaphragm. It is designed for use with food and industrial fluids and slurries that are difficult to measure because of sticking or plugging of orifices. Its hermetically sealed, all stainless steel construction make it suitable for the harshest industrial environments. Ten feet of 2-conductor shielded cable is standard with a second ½ NPT fitting on the body for conduit installation. Pressure ranges from 0 to 50 up to 0 to 750 psi are available to cover most processing and industrial applications.

SPECIFICATIONS

Excitation: 10 to 40 Vdc
Output: 4 to 20 mA 10% adj
Zero Balance: 4 mA +10% -2% adj
Accuracy: 0.5% linearity, hysteresis and repeatability combined
Operating Temp Range: -46 to 121°C (-50 to 250°F)
Compensated Temp Range: 16 to 71°C (60 to 160°F)
Thermal Effects:
Span: 0.003% of rdg/°F
Zero: 0.0045% of FSO/°F
Proof Pressure: 150% of range
Burst Pressure: 300% of range



Dimensions: mm (inch)

Wetted Parts: 17-4 PH stainless steel with 316 stainless steel diaphragm
Pressure Port: ½-14 NPT male
Electrical Connection: 3 m (10') 2-conductor shielded vented cable with ½-14 NPT fitting

To Order Visit omega.com/px43-i for Pricing and Details

RANGE		MODEL NO.	COMPATIBLE METERS*
0 to 50 psig	0 to 3.4 bar	**PX43E0-050GI**	DP41-E, DP25B-E, DP24-E
0 to 60 psig	0 to 4.1 bar	**PX43E0-060GI**	DP41-E, DP25B-E, DP24-E
0 to 100 psig	0 to 6.9 bar	**PX43E0-100GI**	DP41-E, DP25B-E, DP24-E
0 to 200 psig	0 to 13.8 bar	**PX43E0-200GI**	DP41-E, DP25B-E, DP24-E
0 to 300 psig	0 to 20.7 bar	**PX43E0-300GI**	DP41-E, DP25B-E, DP24-E
0 to 500 psig	0 to 34.5 bar	**PX43E0-500GI**	DP41-E, DP25B-E, DP24-E
0 to 750 psig	0 to 51.7 bar	**PX43E0-750GI**	DP41-E, DP25B-E, DP24-E

Comes complete with 5-point NIST traceable calibration.
Metric ranges available – consult Engineering.
** See **omega.com** for compatible meters.*

***Ordering Examples:* PX43E0-100GI,** *100 psi gage model with stainless steel wetted parts, 3 m (10') cable, 4 to 20 mA output.* **PX43E0-050GI,** *50 psi gage model with stainless steel wetted parts, 3 m (10') cable, 4 to 20 mA output.*

B-182

B7. Grundfos Differential Pressure Sensor (reproduced from Ref. 32)

GRUNDFOS DATA SHEET

DPI 0 - 2.5

Differential Pressuresensor, Industry, 0 - 2.5 bar

TM04 4738 1909

Fig. 1 DPI sensor

Technical overview

Grundfos Direct Sensors ™ , type DPI, is a series of differential pressure sensors for industry. The DPI sensors are compatible with wet, aggressive media and are available for differential pressure ranges of 0 - 0.6 up to 0 - 10 bar.

The DPI sensor utilises MEMS sensing technology in combination with a novel packaging concept using corrosion-resistant coating on the MEMS sensing element. This makes the DPI sensor very robust and ideal for pump integration and monitoring in harsh environments.

Applications

- Pump and pump control systems
- Filters (monitoring)
- Cooling and temperature control systems
- Water treatment systems
- Boiler control systems
- Renewable energy systems
- Heat exchanger efficiency (monitoring of fouling).

Features

- Pressure ranges: 0 - 0.6; 0 - 1; 0 - 1.2; 0 - 1.6; 0 - 2.5; 0 - 4; 0 - 6 and 0 - 10 bar differential pressure
- Designed for harsh environments
- Analogue output signal
- Compact and well proven design
- MEMS sensing technology
- Approved for the EU, US and Canadian markets.

Benefits

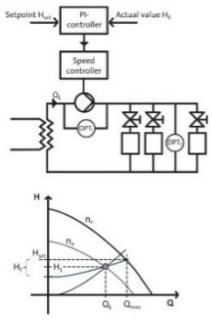
- Compatible with wet, aggressive media
- Accurate, linearised output signal
- Cost-effective and robust design.

Specifications

Pressure	
Measuring range (differential)	2.5 bar
Accuracy (IEC 61298-2)	2 % FS
Response time	< 0.5 s
Static Pressure P_1	16 bar
Static Pressure P_2	10 bar
Max system pressure	16 bar
Media and environment	
Media	Liquids, gasses and air
Media temperature (operation)	–10 to +70 °C
Media temperature (peak)	up to +80 °C
Ambient air temperature	–40 to +70 °C
Ambient air temperature (peak)	–55 to +90 °C
Humidity	0 to 95 % (relative), non-condensing
System burst pressure	25 bar
Electrical data	
Power supply	12-30 VDC
Output signals	4-20 mA
Load impedance	24 V max. 500 kΩ 16 V max. 200 kΩ 12 V max. 100 kΩ
Sensor materials	
Sensing element	Silicon-based MEMS sensor
Seal	FKM rubber
Housing	DIN W.-Nr. 1.4305
Wetted materials	FKM and PPS
Environmental standards	
Enclosure class	IP55
Temperature cycling	IEC 68-2-14
Vibration (non-destructive)	20 to 2000 Hz, 10G, 4h
Immunity	EN 61000-6-2
Emission	EN 61000-6-3
Weight	550 g

Flow compensated differential pressure control

(SPR Reglung)



TM03 0411 5004

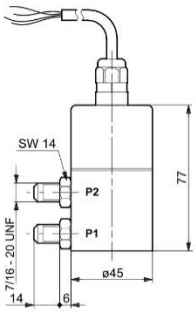
Fig. 2 SPR Reglung

If the equipment is used in a manner not specified by the manufacturer, the protection provided by the equipment may be impaired.

BE > THINK > INNOVATE >

GRUNDFOS

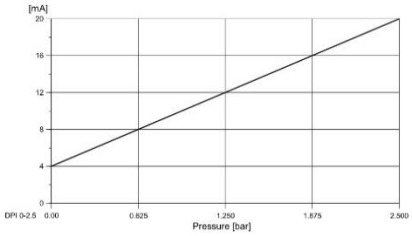
Dimensions [mm]



TM03 2059 3505

Fig. 3 Dimensional sketch

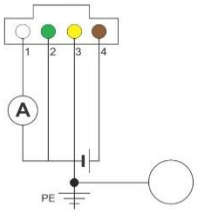
Output signals



TM04 5106 2609

Fig. 4 Differential pressure response

Electrical connections



TM04 3049 3508

Fig. 5 Electrical connections

Pin configuration		Colour
1	Test conductor (can be cut off during mounting). Do not connect this conductor to the voltage supply.	White
2	Signal conductor	Green
3	GND (earth conductor)	Yellow
4	12-30 V supply voltage	Brown

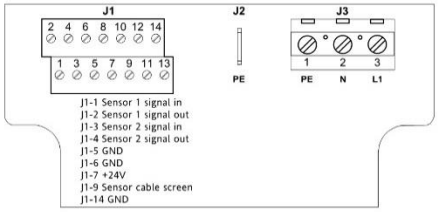
96985463 1109	GB

Sensor Interface type SI 001 PSU

Power supply and amplifier for cables above 30 m and 2 wire connection of 400 VAC

TM04 4194 0809

Fig. 6 Sensor Interface, SI 001 PSU



TM04 4131 0809

Fig. 7 Connections for power supply / amplifier

Part
Sensor Interface, SI 001 PSU

Accessories

Pos.	Component		
A	Fitting 6 mm	AISI 316	Tube connection
	Fitting 8 mm		
	Fitting 6 mm		Cutting ring
	Fitting 8 mm		
B	Cable for DPI 5.0 m		
	Cable for DPI 10.0 m		
	Wall bracket for sensor		

Type key

The DPI sensor is labelled with a type designation.

	96573683	- XX	- XXX	XXXXX
Product number				
Version				
Production year and week				
Consecutive number				

For more information, see
http://www.grundfos.com/directsensors.

Subject to alterations.

Grundfos Sensor A/S
Poul Due Jensens Vej 7. DK-8850 Bjerringbro. Denmark
Telephone: +45 87 50 14 00

www.grundfos.com\directsensors

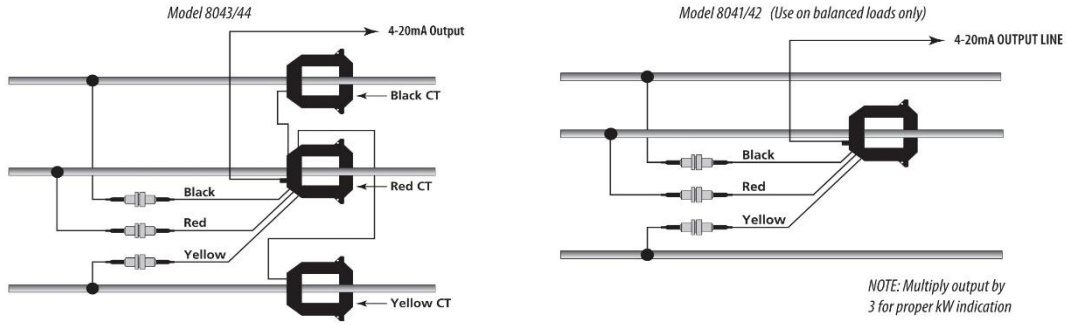
GRUNDFOS

B8. VERIS Power Monitor Sensor (reproduced from Ref. 33)

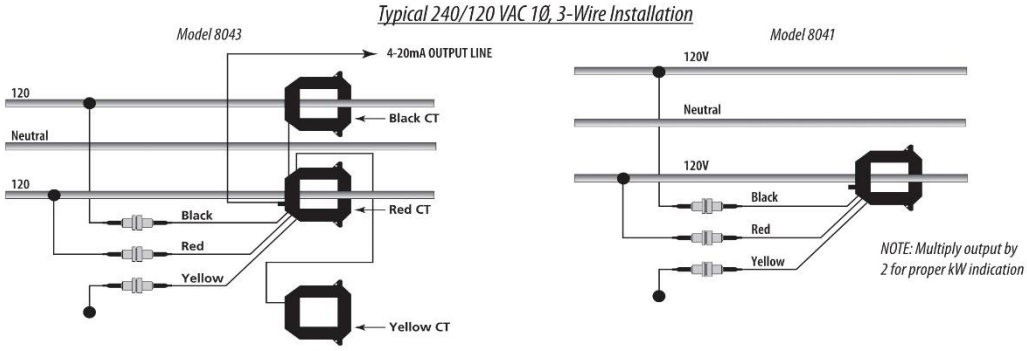
VERIS INDUSTRIES | H804X SERIES | INSTALLATION GUIDE

WIRING

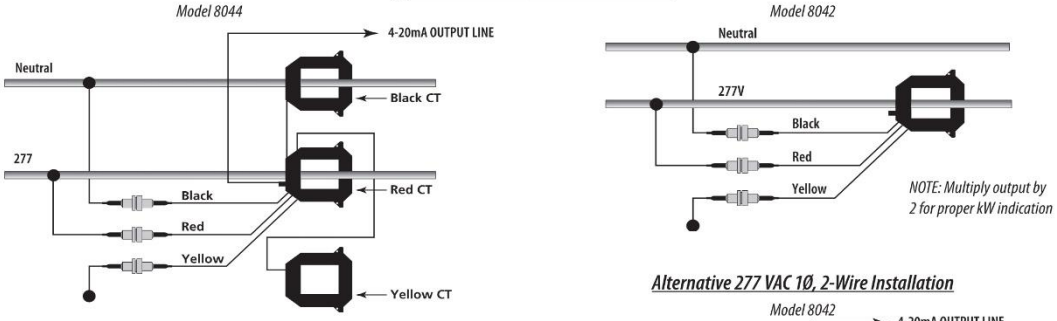
Typical 208/480 VAC 3Ø, 3- or 4-Wire Installation



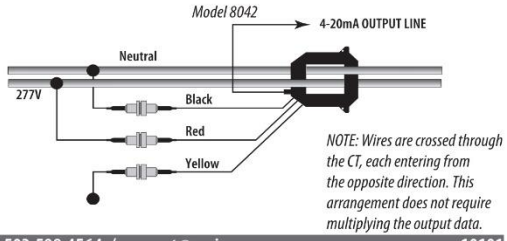
Typical 240/120 VAC 1Ø, 3-Wire Installation



Typical 277 VAC 1Ø, 2-Wire Installation



Alternative 277 VAC 1Ø, 2-Wire Installation



TROUBLESHOOTING

Problem	Solution
Status LED does not blink	Check fuses and voltage connections. Status LED should blink regardless of CTs or output connections.
Readings seem highly inaccurate.	• Check that each CT is installed on the conductor with the corresponding color voltage input lead attached. In most cases, incorrect wiring will cause the STATUS LED to blink RED (slowly). However, a power factor lower than 0.5 could cause the LED to blink this way, even if the unit is installed properly. • It does not matter which side of the CT faces towards the load. • If current is below 7% of full scale maximum for the CT, use a smaller CT or wrap each wire through the CT multiple times • If using the single-phase H8042, use an amp-clamp to ensure that all three phases are passing the same approximate current. If phases are unbalanced, try the H8043/H8044 models.
Meter goes offline when load is switched off.	Voltage leads must be connected on the Line side of the conductor. The power meter cannot communicate without voltage.
Status LED blinks red.	• If the LED blinks quickly (i.e., about 5 blinks in two seconds), then use a higher rated CT. • If the LED blinks slowly (i.e., about 1 blink in two seconds) the CTs are not installed on the correct conductors, or the power factor is less than 0.5. The meter can accurately measure these low PFs, but few loads operate normally at such a low power factor.

NOTES

1. DO NOT GROUND THE SHIELD INSIDE THE ELECTRICAL PANEL. All wires, including the shield should be insulated to prevent accidental contact to high voltage conductors.
2. The cable should be mechanically secured where it enters the electrical panel.
3. The cable should be shielded twisted pair wire BELDEN 1120A or similar.

WARNING: After wiring the cable, remove all scraps of wire or foil shield from the electrical panel. This could be DANGEROUS if wire scraps come into contact with high voltage wires!

MAXIMUM READINGS

Model	3Ø Power (kW)	1Ø Power (kW)
H8041-0100-2	36.03	24.00
H8041-0300-2	108.1	72.00
H8041-0400-3	144.1	96.00
H8041-0800-3	288.2	192.0
H8041-0800-4	288.2	192.0
H8041-1600-4	576.4	384.0
H8041-2400-4	864.6	576.0
H8042-0100-2	83.14	55.43
H8042-0300-2	249.4	166.3
H8042-0400-3	332.6	221.7
H8042-0800-3	665.1	443.4
H8042-0800-4	665.1	443.4
H8042-1600-4	1330	886.7
H8042-2400-4	1995	1330
H8043-0100-2	36.03	36.03
H8043-0300-2	108.1	108.1
H8043-0400-3	144.1	144.1
H8043-0800-3	288.2	288.2
H8043-0800-4	288.2	288.2
H8043-1600-4	576.4	576.4
H8043-2400-4	864.6	864.6
H8044-0100-2	83.14	83.14
H8044-0300-2	249.4	249.4
H8044-0400-3	332.6	332.6
H8044-0800-3	665.1	665.1
H8044-0800-4	665.1	665.1
H8044-1600-4	1330	1330
H8044-2400-4	1995	1995

  USA 800.354.8556 or +1.503.598.4564 / support@veris.com

B9. Mastech HY3003D Power Supply (reproduced from Ref. 34)

Power Supply *HY30XX Series*

MODEL EXPLAINATION:

HY XX XX X - X

① ② ③ ④ ⑤

① Products of MASTEC
② Output voltage numbers
③ Output current numbers
④ no: LED display
D: LCD display
C: two pointer meters display
S: four pointer meters display
⑤ no: single output voltage current regulated
2: double output voltage current regulated
3: double output voltage current regulated + fixed 5V3A

MODEM	HY3002D	HY3003D	HY3005D
Input Voltage	110/220V±10%		
Output Voltage	0 ~ 30V		
Output Current	0 ~2A	0 ~ 3A	0 ~ 5A
Source Effect	CV ≤ 0.01%±1mV; CC ≤ 0.02%±1mA		
Load Effect	CV ≤ 0.01%±5mV; CC ≤ 0.02%±5mA		
Ripple & Noise	≤ 1mVrms		
Display	Two 3 1/2 digit LCD display		
Accuracy	V: ±1%±2digits; C: ±2%±2digits		
Size	291 X 158 X 136mm		
Weight	3 ~ 6kg		

MODEM	HY3002C	HY3003C	HY3005C
Input Voltage	110/220V ± 10%AC		
Output Voltage	0 ~ 30V		
Output Current	0 ~ 2A	0 ~ 3A	0 ~ 5A
Source Effect	CV≤ 0.01%±1mV; CC≤ 0.02%±1mA		
Load Effect	CV≤ 0.01%±5mV; CC≤ 0.02%±5mA		
Ripple & Noise	≤ 1mVrms		
Display	Voltage & Amperometer display		
Accuracy	2.5%f.s.		
Size	291 X 158 X 136mm		
Weight	3 ~ 6kg		

Sinotech Shanghai
Room 1404, 1759 North Zhongshan Road, Putuo District,
Shanghai, China

Page 1

B10. Suresite Level Transducer and Visual Indicator (reproduced from Ref. 35)

Standard Alloy Versions – Standard Size

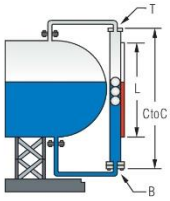
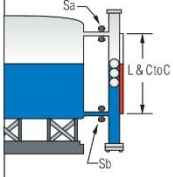
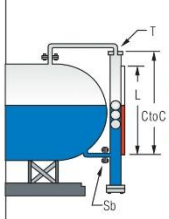
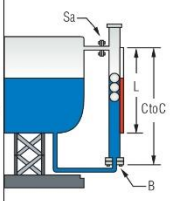
ORDER IT!
Ordering is Easy! See Page D-9.
Easy online ordering too!

- Temperatures to 750°F (399°C)
- Pressures to 700 PSI (48 bar)

Rugged, welded construction makes these 2-1/2″ (63.5 mm) diameter design, alloy SureSite Indicators dependable over a long service life indoors and out.

1. Mounting Configuration Types

To choose the best configuration for your application, focus on the process connections (connections where the liquid typically enters/leaves the SureSite).

	Type AA Top and Bottom Process Connections	Type BA Side and Side Process Connections	Type CA Top and Side Process Connections	Type DA Side and Bottom Process Connections
L = Length of Visual Indication				
Typical Lengths*	C to C = L + 10-1/4″ (260.4 mm)	C to C = L	C to C = L + 3-3/4″ (95.2 mm)	C to C = L + 6-1/2″ (165.1 mm)
Flag Material	Plastic (300°F/148.9°C) or Aluminum (750°F/399°C)			
Length of Indication (Uninterrupted)	240″ (610 cm)			
Minimum Specific Gravity	0.39			

* Dimensions vary due to connections, material and specific gravity.
Note: Additional materials, floats, connections and manufacturing techniques are available to extend lengths and operational capabilities.
Please contact GEMS Sensors if the parameters above do not meet your requirements.

Type BA Shown

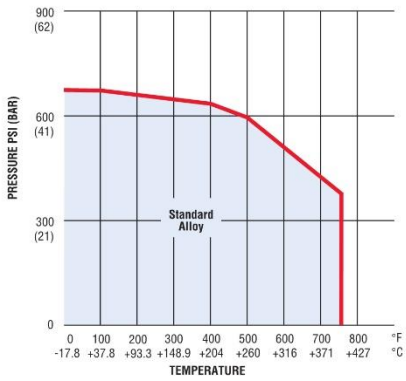
2. Material

Housing and Float: 316 Stainless Steel
Pressure/Temperature performance parameters for alloy SureSite versions are specified in the chart at right.
Please consult the factory with temperature/pressure requirements that fall outside the parameters shown here.

= Stock Material (Best economy and delivery).

Materials		Code
Housing	Float	
316L Stainless Steel	316L Stainless Steel	2
Carpenter 20	Hastelloy C276	3*
Hastelloy C276	Hastelloy C276	4*

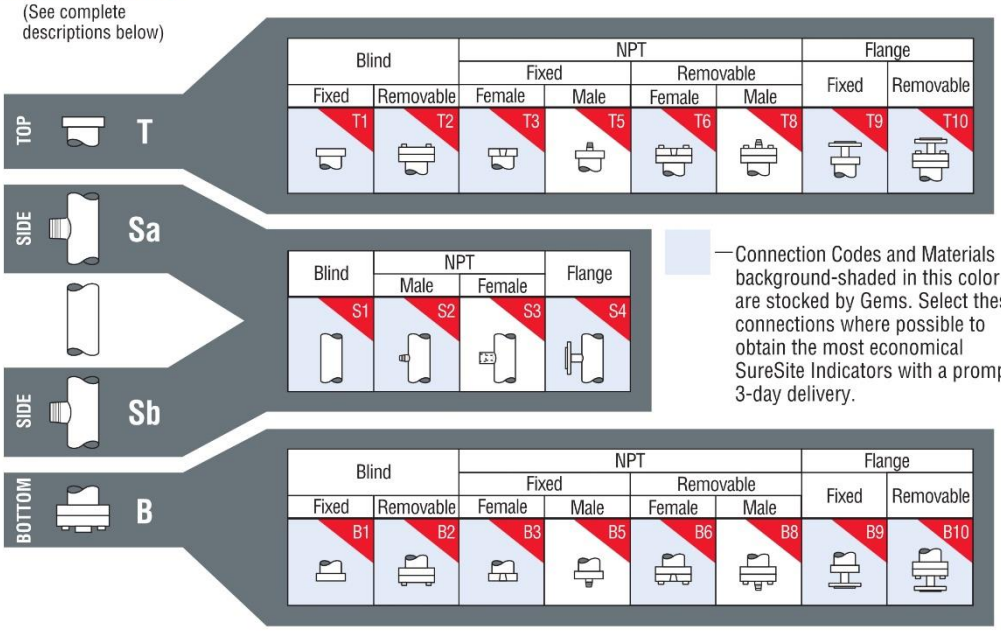
* Consult factory for pressure/temperature capabilities.



Note: SureSite Indicators are available for temperatures as low as -200°F (-129°C).

3. Connection Codes

(See complete descriptions below)



Connection Codes and Materials background-shaded in this color are stocked by Gems. Select these connections where possible to obtain the most economical SureSite Indicators with a prompt 3-day delivery.

Connection Code Descriptions
Please provide all connections when completing the OrderIt! Product Check List (located on the following page).
Note: Before selecting your connections, consider incorporating your vent and drain requirements.

T & B (Top and Bottom)

T/B 1. Welded pipe cap
T/B 2. Standard fixed flange/blind mating flange
T/B 3. Welded pipe cap w/FNPT
T/B 5. Welded pipe cap w/MNPT nipple
T/B 6. Standard fixed flange/mating FNPT reducing flange
T/B 8. Standard fixed flange/mating flange with MNPT nipple
T/B 9. Welded pipe cap with ANSI flange
T/B 10. Standard fixed flange/mating reducing flange spool

Sa & Sb Sides

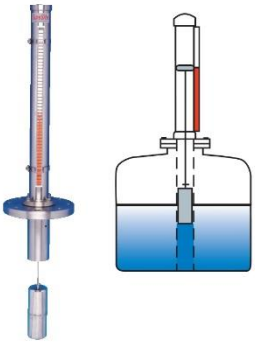
S1. No connection
S2. MNPT nipple
S3. FNPT coupling
S4. ANSI flange

3-Day Qwik Configured
Available for up to 10 units
Built & Shipped in 3 Days!

Need it quick? Choose materials and components with the color shading for 3-Day manufacturing and shipping. See the Product Configurator section at www.gemssensors.com for further details.

Top Mount Units

When it's not practical to access the side of a tank for liquid monitoring, look to SureSite Top Mount Indicators for the solution. Please consult with the factory for these specially configured indicators **1-800-378-1600.**



Accessories – Pages D-16 to D-18

Make more of your SureSite® Indicator with the productivity-enhancing accessories found at the end of this section.

- **Indicating Scales**
  Add graduations to your flag indication.
- **Switch Modules**
  Control pumps, valves, alarms, etc. Mount externally on housing for infinite positioning.
- **Continuous Output Transmitters**
  Signal conditioned for compatibility with most electronic instruments to 300°F (149°C).

B11. HOBO UX120-006M Data Logger (reproduced from Ref. 36)

onset®
HOBO® Data Loggers

HOBO UX120 4-Channel Analog Logger

Flexible, Accurate, 4-channel Analog Logger

The HOBO UX120-006M Analog Logger is a high-performance, LCD display data logger for building performance monitoring applications.

As Onset's highest-accuracy data logger, it provides twice the accuracy of previous models, a deployment-friendly LCD, and flexible support up to four external sensors for measuring temperature, current, CO2, voltage and more.

Supported Measurements: Temperature, 4-20mA, AC Current, AC Voltage, Air Velocity, Carbon Dioxide, Compressed Air Flow, DC Current, DC Voltage, Gauge Pressure, Kilowatts, Volatile Organic Compound (sensors sold separately)

Key Advantages:

- Twice the accuracy over previous models
- 16-bit resolution
- Flexible support for a wide range of external sensors
- LCD confirms logger operation and displays near real-time measurement data
- Provides minimum, maximum, average and standard deviation logging options
- On-screen alarms notify you when a sensor reading exceeds set thresholds
- Stores 1.9 million measurements for longer deployments between offloads

Minimum System Requirements:

Software USB cable*

For complete information and accessories, please visit: **www.onsetcomp.com**

Part number	**UX120-006M** (4-Channel Analog)
Memory	1.9 Million
Logging Rate	1 second to 18 hours, user selectable
Logging Modes	Normal, Burst, Statistics
Memory Modes	Wrap when full or stop when full
Time Accuracy	±1 minute per month at 25°C (77°F)
Battery Life	1 year typical with logging rate of 1 minute and sampling interval of 15 seconds or greater, user replaceable, 2 AAA
Dimensions	10.8 x 5.41 x 2.54 cm (4.25 x 2.13 x 1 in.)
Operating Range	Logging: -20° to 70°C (-4° to 158°F); 0 to 95% RH (non-condensing)
Accuracy	±0.1 mV ±0.1% of reading
CE Compliant	Yes

Contact Us

Sales (8am to 5pm ET, Monday through Friday)
- Email sales@onsetcomp.com
- Call 1-508-759-9500
- In U.S. toll free 1-800-564-4377
- Fax 1-508-759-9100

Technical Support (8am to 8pm ET, Monday through Friday)
- Contact Product Support onsetcomp.com/support/contact
- Call 1-508-759-9500
- In U.S. toll free 1-877-564-4377

Onset Computer Corporation
470 MacArthur Boulevard
Bourne, MA 02532

B12. Gateway LT2802u Laptop

This laptop served as the main data acquisition system for the recording of data in the power plant. The HOBOWARE software installed on this laptop was used to record the data for time intervals of one minute each.

The laptop contains a hard disk drive of 250 GB and a RAM of 1 GB. The screen is 10 inch LED LCD display. The laptop operates on Li-ion batteries.

Appendix C: Calibration of HOBO TMC6-HE Temperature Sensors

To calibrate the HOBO TMC6-HE temperature sensors, the sensors were placed in an ice-water bath. Later the sensors were placed in a vessel of water, which was stirred. This was done to determine the accuracy of the sensors when placed in direct contact with a common medium (e.g., water) at a known temperature. The differences among the temperature data were then recorded; and, based on these differences, it was then determined if the temperature sensors should be used or not for the measurement of temperatures in the heat exchangers. As can be seen in Fig. C1, the temperature sensors were first put in an ice-water bath, while Fig. C2 shows an example of real time temperature data from the HOBO data logger.

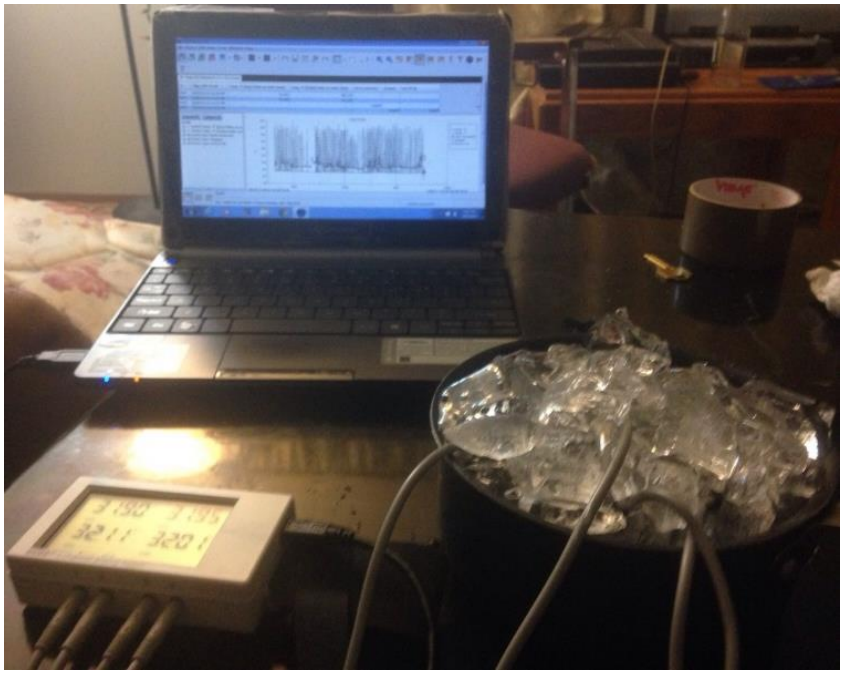


Fig. C1. Four temperature sensors in ice-water bath

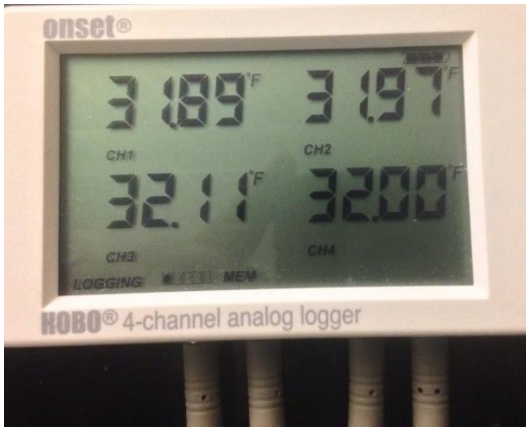


Fig. C2. HOBO data logger displaying real time temperatures of the four sensors in ice-water bath

The temperature of each sensor was plotted vs. time, as shown in Fig. C3. It can be seen that the sensors measured the temperature of ice (i.e., 32° F). Also, the sensors were very accurate when compared to each other (see Fig. C2 for one example).

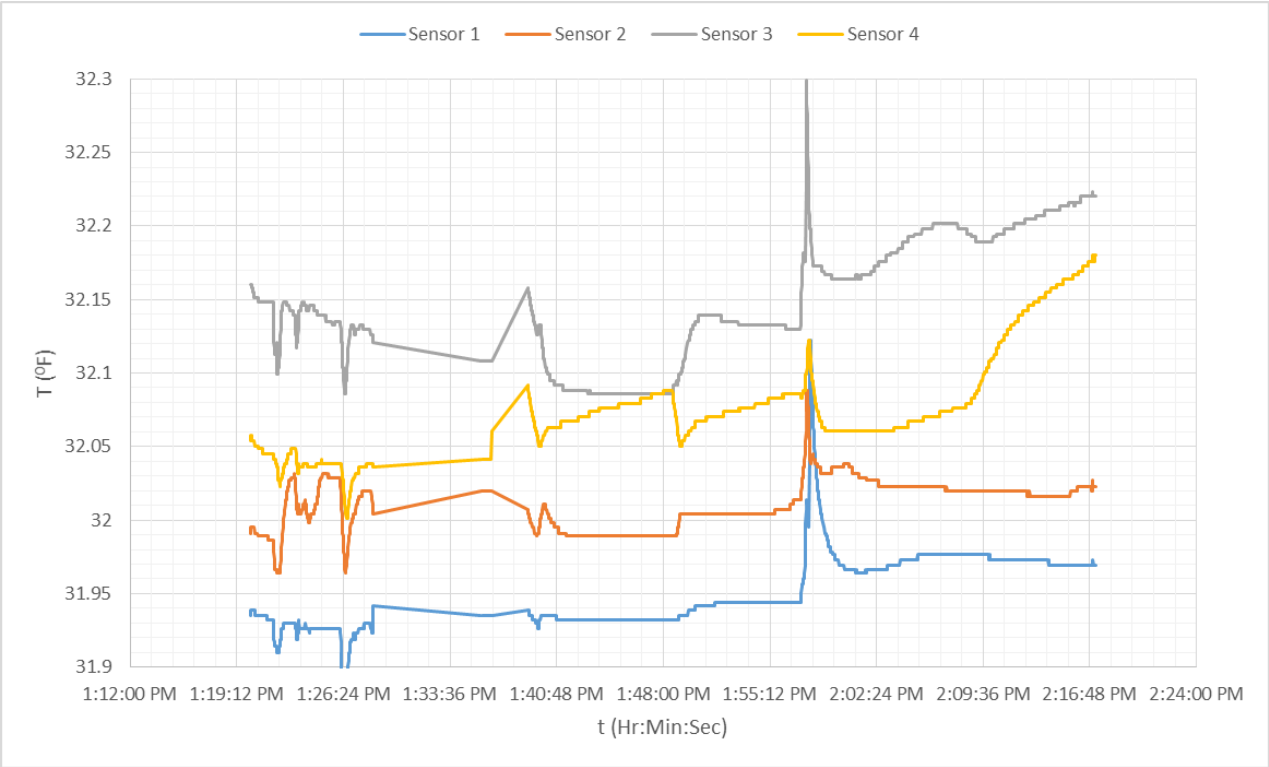


Fig. C3. Plot of the temperatures of the four sensors while submerged in an ice-water bath

The average and the corresponding standard deviation for each sensor are tabulated in Table C1.

Table C1. Averages and standard deviations of the temperature sensors' outputs when placed in ice-water bath

Sensor No.	Average (°F)	Standard Deviation (°F)
1	31.95	0.006
2	32.01	0.018
3	32.14	0.011
4	32.07	0.006

However, since the power plant has hot water flowing in the heat exchanger, the sensors were also tested in warm water. Water was heated to approximately 103.5° F and then stirred continuously. The sensors were submerged in the heated water before the water was stirred (see Fig. C4). The water was stirred continuously for the entire time of the test to best minimize temperature gradients in the water, as shown

in Fig. C5. The real time temperatures of the four sensors in hot water (i.e., when stirred) are shown in Fig. C6.

Fig. C4. The four temperature sensors submerged in hot water (standing water)

Fig. C5. The four temperature sensors submerged in hot water (stirred continuously)

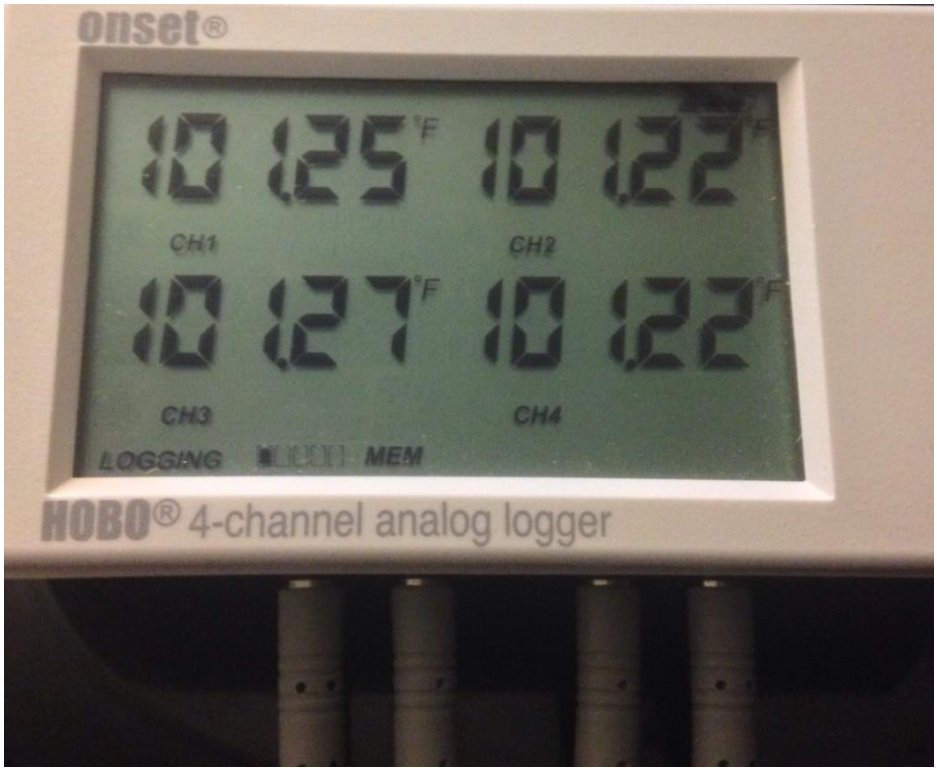


Fig. C6. HOBO data logger displaying real time temperatures of the four sensors in hot water (stirred)

The water was then heated to approximately 160 ^{o}F; and the temperature recording process was repeated. The resultant temperatures were plotted against time as can be seen in Fig. C7. The temperature sensors logged temperature data whose values were extremely close to each other. When the temperatures of the four sensors were compared, with sensor 1 as the reference, the maximum error among them was approximately ± 0.02% (see Fig. C8). This error was considered to be acceptable; and subsequently, the four sensors were installed on the inlet and outlet pipelines of the heat exchanger in the basement.

The sensors for the vent condenser on the first floor were also calibrated, and the maximum error among them was also approximately ± 0.02%. The sensors were installed on the surface of the vent condenser tubing, and then temperatures for the condensate water flowing in the vent condenser tubing were logged, as shown in Figs. C9 and C10. It can be seen that the temperature differences (Fig. C10) between the inlet and outlet water to/from the vent condenser were in the range of 9.4-9.7 ^{0}F. The average temperature difference was 9.6 ^{0}F while the standard deviation was 0.055 ^{0}F.

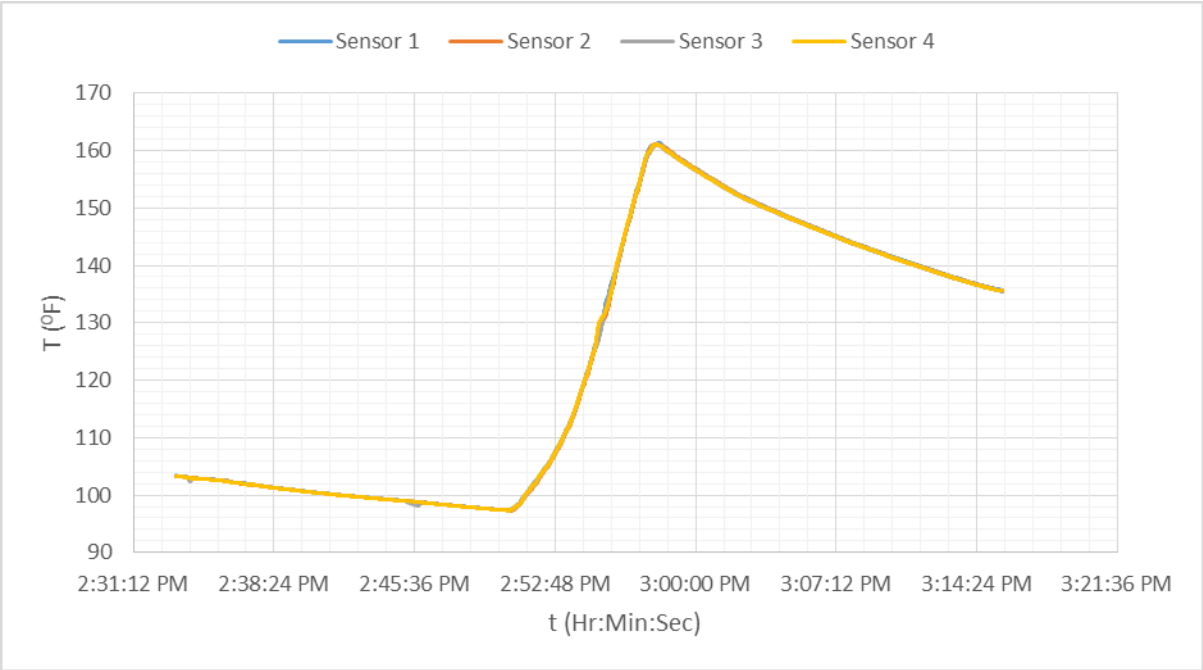


Fig. C7. Plot of the four temperature sensors' outputs while submerged in hot water

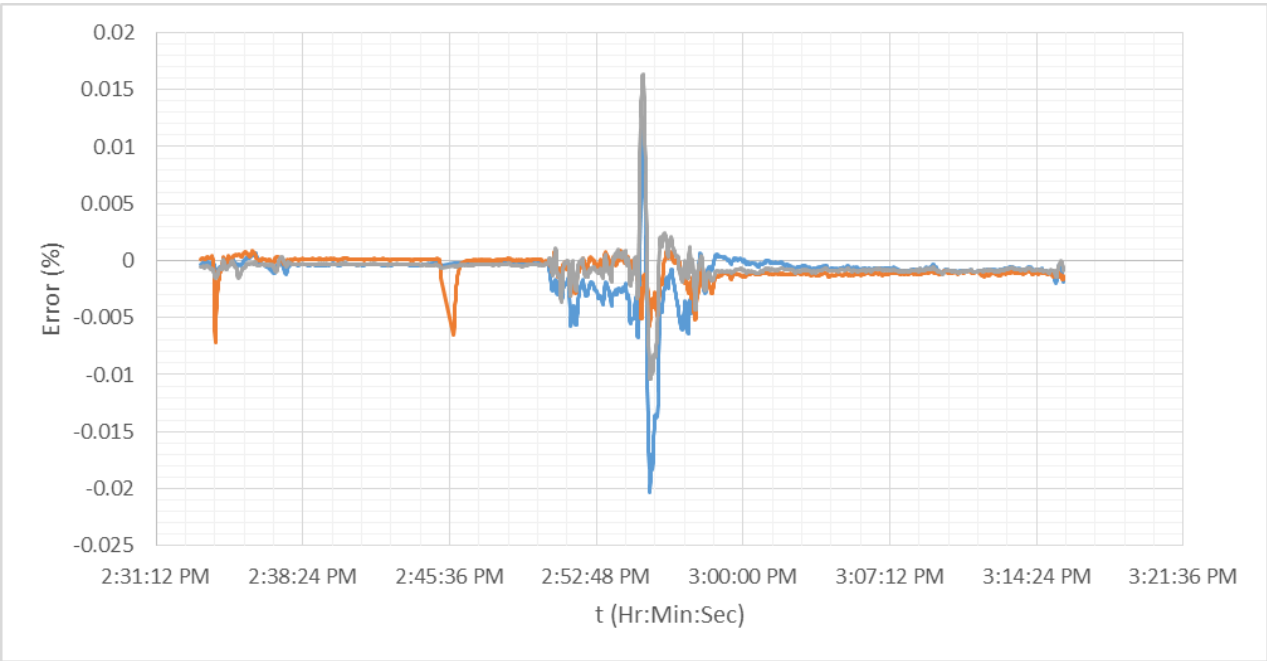


Fig. C8. Errors of three of the temperature sensors in warm water as compared to the fourth sensor

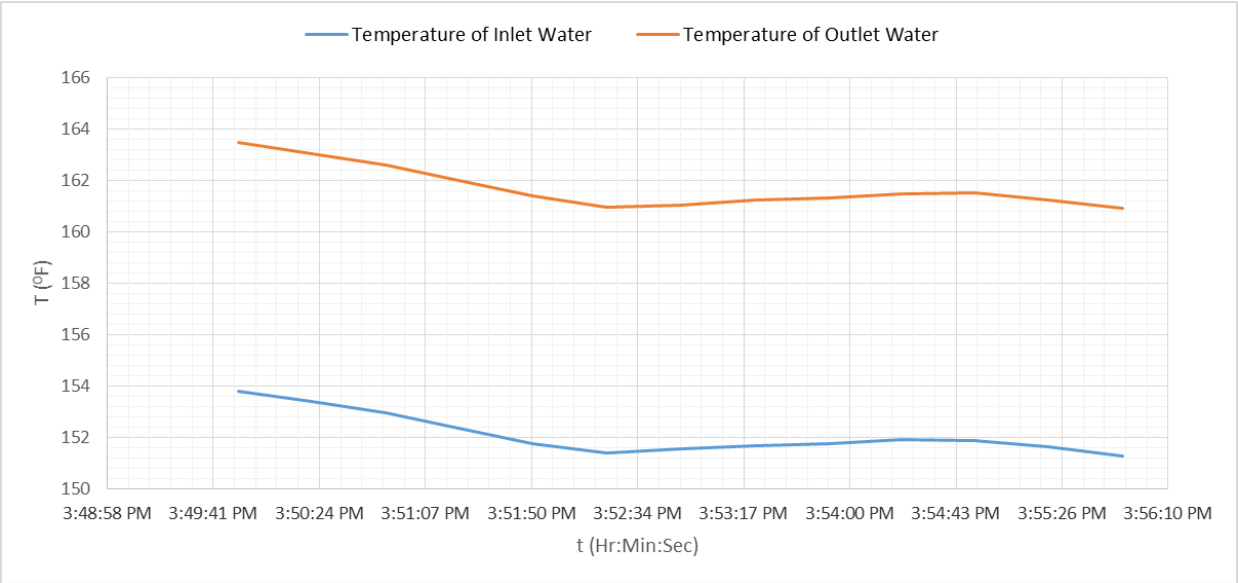


Fig. C9. Temperatures of water at the inlet and outlet of vent condenser

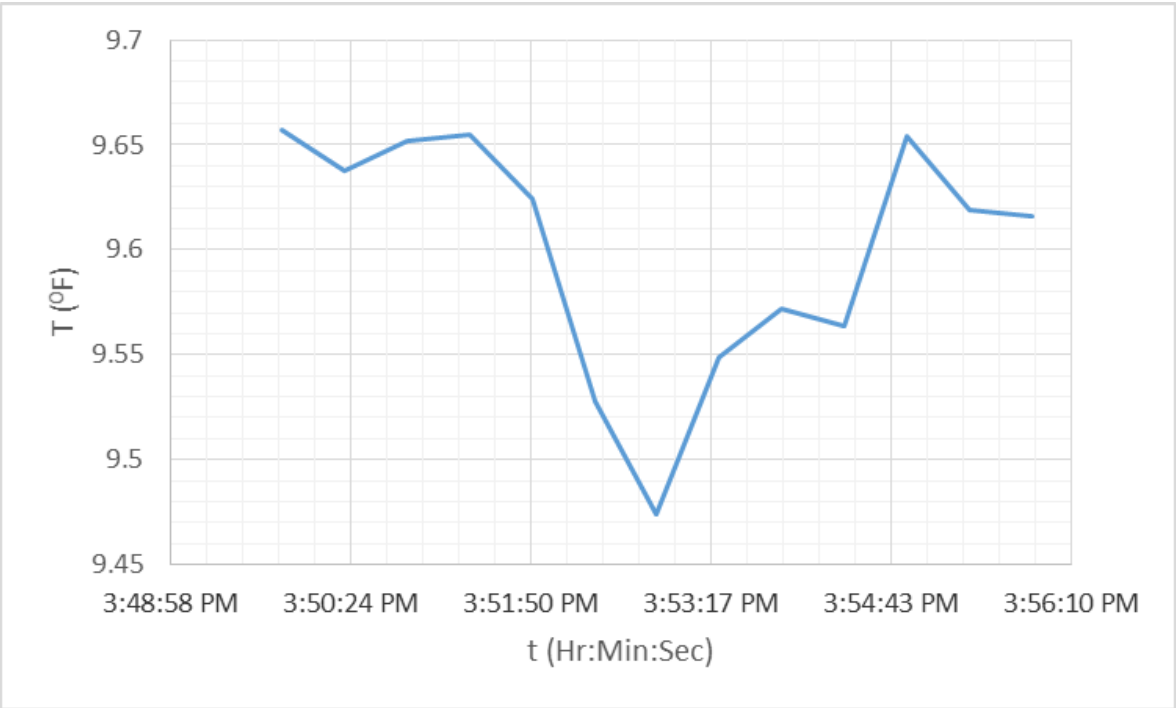


Fig. C10. Temperature difference between the outlet and inlet of vent condenser

The sensors were then interchanged between the inlet and outlet in order to check the consistency. The resulting temperature plots and differences are shown in Figs. C11 and C12. After the interchange of the sensors, the average temperature difference was 9.96 ^{0}F with a standard deviation of 0.057 ^{0}F.

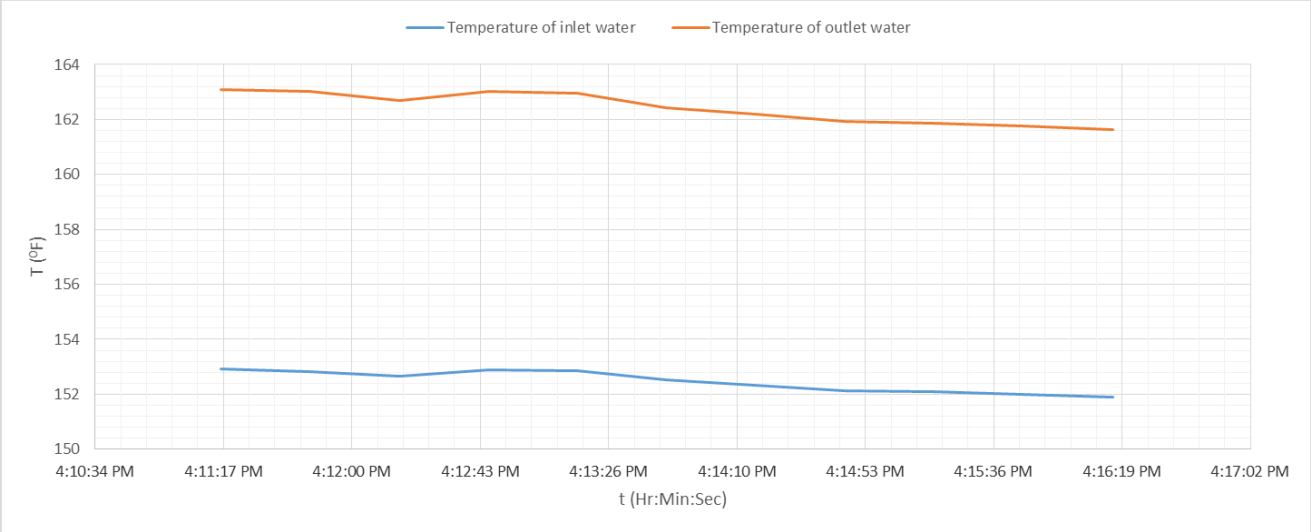


Fig. C11. Temperatures of water at the inlet and outlet of vent condenser (after interchanging sensors)

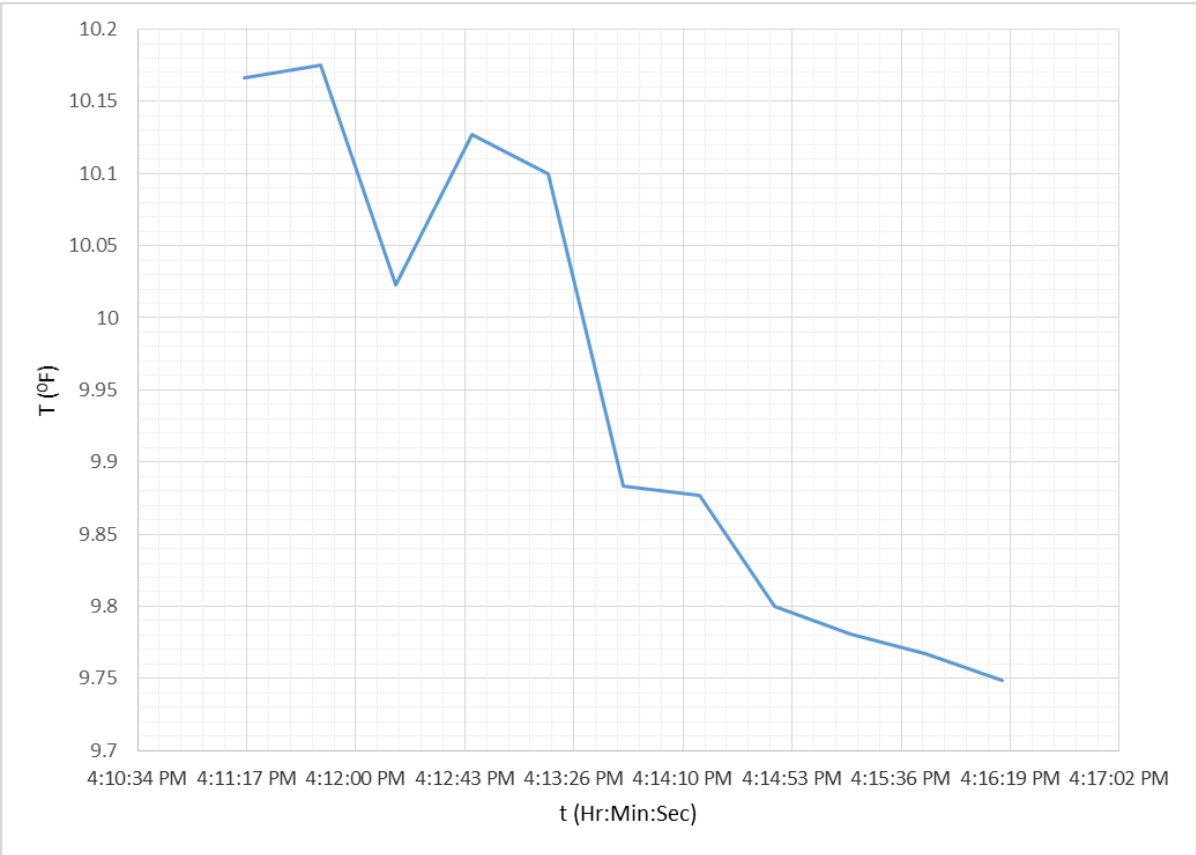


Fig. C12. Temperature difference between the outlet and inlet of vent condenser (after interchanging sensors)

Also, the temperatures shown by both analog temperature gauges at the inlet and at the outlet of the vent condenser were recorded manually. This was done for 5 minutes each at a time interval of 30 seconds (see Table C2). The data was collected before and after switching the digital sensors.

From this data, it can be seen that, for both cases (i.e., before and after the switching of the sensors), the inlet temperatures from the two analog gauges were close to the digital sensor temperatures in the plots shown in Fig. C9 and Fig. C11. The outlet temperatures from the analog gauges were as close to those of the digital sensors as for the inlet temperatures. The outlet temperatures of the digital sensors were 6-9 ^{o}F less than those of the analog gauge outlet temperatures. This difference of 6-9 ^{o}F was consistent throughout the entire time interval (i.e., for the entire 5 minutes). Aging of the analog temperature gauges might have caused inaccuracies in temperature readings. The digital sensors were calibrated and tested extensively for both the basement heat exchanger and the vent condenser. The maximum percentage error was only ± 0.02% for the digital sensors. Also, the digital sensors logged temperature data continuously for every minute. There was also no need to buy a separate data acquisition system for the digital sensors because the existing HOBOWARE software could be used for recording and plotting data. For these reasons, the data from the digital temperature sensors was used in this thesis.

Table C2. Manually recorded temperatures by looking at analog temperature gauges for 30 second intervals

Before interchanging digital sensors			After interchanging digital sensors		
Time	Temp. In (^{0}F)	Temp. Out (^{0}F)	Time	Temp. In (^{0}F)	Temp. Out (^{0}F)
3:50:00	153	170	4:11:00	154	169
3:50:30	152	169	4:11:30	154	169
3:51:00	152	168	4:12:00	154	169
3:51:30	152	168	4:12:30	154	169
3:52:00	152	167	4:13:00	154	169
3:52:30	152	167	4:13:30	154	169
3:53:00	152	167	4:14:00	154	169
3:53:30	152	168	4:14:30	154	168
3:54:00	152	168	4:15:00	154	168
3:54:30	152	168	4:15:30	154	168
3:55:00	152	168	4:16:00	154	168
3:55:30	152	167	4:16:30	154	168

Appendix D: HOBOWARE Linear Scaling Assistant Window

The Siemens flow meters send 4-20 mA output signals. The HOBO data logger receives the 4-20 mA signals from the Siemens flow meters and then converts them into the desired output values in gpm. This is done by Linear Scaling. From Fig. D1, the "raw" values are the 4-20 mA signals sent by the Siemens flow meters. According to the HOBOWARE software, the raw values have to be set so that Value 1 is the minimum while the Value 2 is the maximum. Thus, Value 1 is set at 4 mA while Value 2 is set at 20 mA. The "scaled" values are then correspondingly set according to the device's minimum and maximum measurement values. A sample screenshot of the linear assistant window for the Siemens flow meter is shown in Fig. D1. Here, the window shows the default scaled values. Initially, they are 0 for Value 1 and Value 2. These values were later changed, scaled Value 1 was set at 0 while scaled Value 2 was set at 528.3 gpm (as the minimum and maximum flow rate of the flow meter were 0 and 528.3 gpm, respectively).

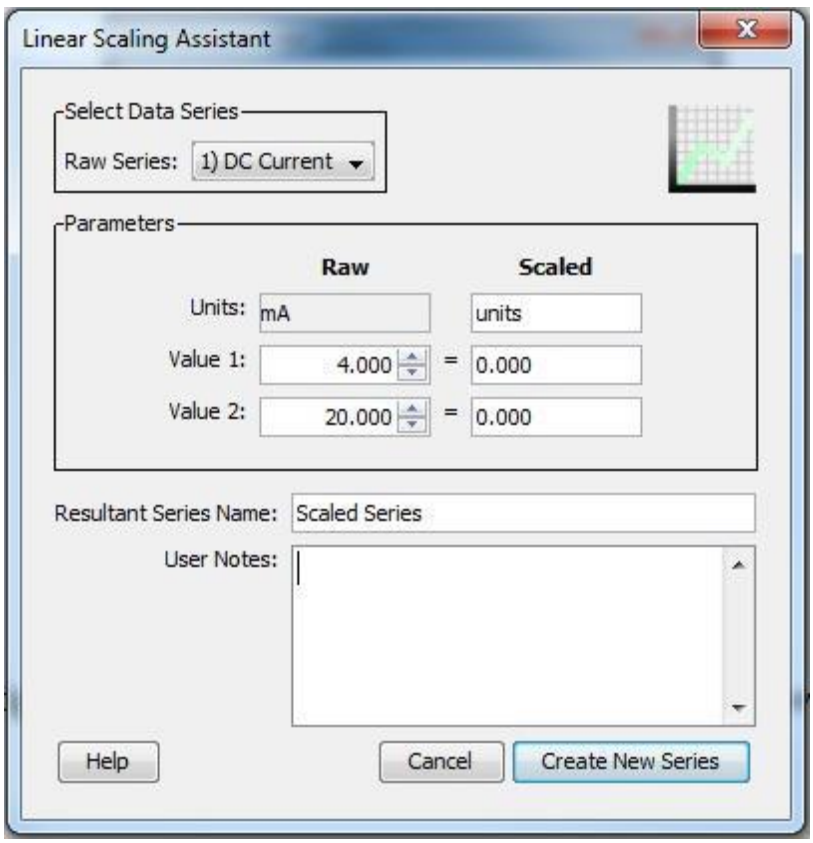


Fig. D1. Screenshot of HOBOWARE Linear Scaling Assistant Window

Once these data values were logged, HOBOWARE plotted the data values vs. their respective recording times. HOBOWARE was set so that data was recorded every minute. In this way, the discharge pressure, flow rate and power consumption data of the Worthington and Grundfos pumps, and the temperature data from the temperature sensors were recorded on a per-minute basis.

Appendix E: Boiler and Plant Efficiency Calculations

The power plant had log sheets displaying boiler efficiency, however there was no explanation of how it was determined. So, using Eq. (3b), boiler efficiency was calculated.

Examples of the log sheets from the power plant displaying boiler efficiency (right most column), the volume of natural gas used, and the mass of steam generated by the boilers are shown in Figs. E1 and E2.

Makeup Water	%	OCT 2015	#1	#2	#7	#8	HRS. on OIL	OIL Gals.	GAS FEET M. No. 1	GAS FEET M. No. 2	GAS FEET M. No. 7	GAS FEET M. No. 8	Steam Lbs. No. 1	Steam Lbs. No. 2	Steam Lbs. No. 7	Steam Lbs. No. 8	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.
20,540	30.5	1			x	x					560,600	6,900			547,300	11,400	558,700		558,700	78
17,010	24.7	2			x						593,800				572,300		572,300		572,300	82
14,260	21.7	3			x						563,800				546,400		546,400		546,400	82
13,540	20.7	4			x						555,000				542,900		542,900		542,900	82
10,870	15.8	5			x						586,300				572,700		572,700		572,700	82
11,140	16.1	6			x						585,600				573,600		573,600		573,600	82
11,440	16.9	7			x						575,000				562,600		562,600		562,600	82
9,320	15.1	8			x	x					248,800	333,100			242,800	269,100	511,900		511,900	82
10,165	14.6	9				x						675,600				576,200	576,200		576,200	82
11,065	16.5	10				x						655,000				557,400	557,400		557,400	82
9,605	16.8	11				x						564,400				473,800	473,800		473,800	82
10,645	19.2	12				x						535,600				459,400	459,400		459,400	82
13,500	22.3	13				x						598,800				503,100	503,100		503,100	82
14,750	23.1	14				x						633,100				529,400	529,400		529,400	82
15,800	23.1	15				x						667,500				567,900	567,900		567,900	83
15,600	20.3	16				x						741,300				639,200	639,200		639,200	83
17,565	23.2	17				x						731,900				627,400	627,400		627,400	83
13,505	20.2	18				x						651,900				554,300	554,300		554,300	83
12,080	19.0	19				x						616,300				528,900	528,900		528,900	83
9,890	16.5	20				x						579,400				496,100	496,100		496,100	83
9,960	16.7	21				x						581,300				495,300	495,300		495,300	83
9,890	15.4	22				x						627,500				532,900	532,900		532,900	83
9,580	14.9	23				x						627,500				533,800	533,800		533,800	83
14,135	20.5	24				x						665,600				571,400	571,400		571,400	83
18,685	25.0	25				x						731,900				619,900	619,900		619,900	83
19,340	23.6	26				x						809,400				681,600	681,600		681,600	83
16,360	19.9	27				x						806,900				681,608	681,608		681,608	83
13,880	16.9	28				x						808,100				681,600	681,600		681,600	83
16,625	16.6	29				x						988,800				833,400	833,400		833,400	83
15,455	15.3	30				x						989,400				840,900	840,900		840,900	83
13,800	16.6	31				x						805,600				691,700	691,700		691,700	83
420,000	19.2								0	0	4,268,900	16,432,800	0	0	4,160,600	13,957,708	18,118,308	0	18,118,308	

Fig. E1. Log sheet from power plant displaying boiler efficiency, volume of natural gas used, and mass of steam produced for October of 2015

In order to use Eq. (2) for energy gain calculation, the volume flow rate of the condensate water to the vent condenser ($Q_{vent/HEX}$) and that of the makeup water to the basement heat exchanger ($Q_{vent/HEX}$) have to be converted into mass flow rate. (Note that $Q_{vent/HEX}$ is used for either the vent condenser or the makeup water flow rate.) This is done by multiplying the volume flow rate by the density of water (ρ). The energy gain of the condensate and makeup water is calculated by multiplying the respective mass flow rates by the specific heat capacity of water (c_p) and the temperature rise of the water (ΔT_{rise}) across the vent condenser and basement heat exchanger respectively (see Eq. (2)). Water enters the economizer as compressed liquid at around 225 ^{0}F and it is converted into saturated steam in the boiler at approximately 375 ^{0}F. This change in enthalpy has to be taken into account in the energy gain calculations.

Once the energy gain is known, the boiler efficiency can be calculated by dividing the energy gain by the volume of gas used in the boiler multiplied by the LHV_{fuel} of natural gas, using Eq. (3b).

Makeup Water	%	NOV 2015	#1	#2	#7	#8	HRS. on OIL	OIL Gals.	GAS FEET M. No. 1	No. 2	No. 7	No. 8	Steam Lbs. No. 1	No. 2	No. 7	No. 8	By GAS STEAM LBS.	By OIL STEAM LBS.	TOTAL STEAM GENERATED	BOILER EFF.
12,490	16.3	1			x							742,500				635,000	635,000		635,000	83
14,220	18.9	2			x							743,800				623,500	623,500		623,500	82
13,590	16.3	3			x							806,900				691,700	691,700		691,700	82
10,350	13.3	4			x	x					391,900	314,400			377,100	268,200	645,300		645,300	82
12,300	15.7	5			x						673,100				652,300		652,300		652,300	83
18,160	20.2	6			x						773,800				745,600		745,600		745,600	83
16,590	18.9	7			x						749,400				729,300		729,300		729,300	83
18,800	20.7	8			x						780,000				755,100		755,100		755,100	83
24,760	24.1	9	x	x	x	x				15,800	800,000	110,600			767,800	84,400	852,200		852,200	83
14,110	15.6	10			x						777,500				752,900		752,900		752,900	83
13,790	16.4	11			x						720,000				698,700		698,700		698,700	83
18,180	18.2	12			x						857,500				827,800		827,800		827,800	83
22,750	22.0	13	x		x				104,400		796,300		83,100		773,900		857,000		857,000	83
22,610	25.5	14			x						756,300				735,000		735,000		735,000	83
19,030	21.6	15			x						752,500				732,800		732,800		732,800	83
14,705	14.9	16			x						858,100				817,700		817,700		817,700	83
15,555	17.1	17			x						792,500				756,900		756,900		756,900	83
21,060	19.1	18	x		x				181,900		806,300		149,600		764,800		914,400		914,400	83
23,085	18.9	19	x		x				318,800		748,800		280,900		731,100		1,012,000		1,012,000	83
21,275	17.8	20	x		x				338,800		706,300		298,800		692,100		990,900		990,900	83
22,070	16.0	21	x		x				527,500		682,500		465,900		676,800		1,142,700		1,142,700	82
22,320	17.1	22	x		x				495,000		655,600		437,100		649,300		1,086,400		1,086,400	82
23,020	19.3	23	x		x				406,300		640,600		358,800		631,300		990,100		990,100	82
20,900	19.2	24	x		x				338,100		616,900		298,400		606,400		904,800		904,800	83
16,930	18.9	25	x		x				21,300		741,900		19,300		722,300		741,600		741,600	83
26,590	25.5	26	x		x				228,800		685,600		194,700		672,400		867,100		867,100	83
29,880	20.7	27	x		x				593,100		684,400		520,600		679,900		1,200,500		1,200,500	83
24,360	16.2	28	x		x				678,100		655,100		593,700		651,400		1,245,100		1,245,100	83
23,500	16.0	29	x		x				670,600		633,800		588,000		631,300		1,219,300		1,219,300	83
24,790	16.2	30	x		x				695,600		669,400		606,400		660,200		1,266,600		1,266,600	83
	#####	31															0		0	
581,770	18.5								5,598,300	15,800	19,406,100	2,718,200	4,895,300	0	18,892,200	2,302,800	26,090,300	0	26,090,300	

Fig. E2. Log sheet from power plant displaying boiler efficiency, mass of natural gas used, and mass of steam produced for November of 2015

Using Eq. (3b) from Section 3.i, the boiler efficiency, η_{boiler}, for October 5, 2015, can be calculated. The following inputs, obtained from Fig. E1, were used

$$Vol_{fuel,day} = 586{,}300\ \frac{ft^3}{day},\ \ m_{steam,day} = 572{,}700\ \frac{lb_m}{day}$$

The average per-minute condensate flow into the vent condenser was $Q_{vent/HEX} = 87.25\ gpm$ (taking the average $Q_{vent/HEX}$ from Table F1), and the average per-minute makeup water flow into the basement heat exchanger was $Q_{vent/HEX} = 14.48\ gpm$ (taking the average $Q_{vent/HEX}$ from Table F2). Using water density at 160 ^{0}F, $\rho = 61\ lb_m/ft^3$; average temperature rise of condensate water, $\Delta T_{rise} = 8.31$ °F (taking the average ΔT_{rise} from Table F1); average temperature rise of makeup water, $\Delta T_{rise} = 17.06$ °F (taking the average ΔT_{rise} from Table F2); specific heat of water at 160 ^{0}F, $c_p = 1.0004\ (Btu/(lb_m\ °F))$; and conversion factors, $C_2 = 0.00228\frac{ft^3}{s\ gpm}$ and $C_9 = 86400\frac{s}{day}$, η_{boiler} can be calculated. Putting the required values into Eq. (3b) gives

$$\eta_{boiler} = \frac{\left(\left(572700\ \frac{lb_m}{day}\right)\left(1197.7\ \frac{Btu}{lb_m}\right) - \left(\frac{572700}{0.98}\ \frac{lb_m}{day}\right)(193.3\ \frac{Btu}{lb_m})\right)}{(586300\ \frac{ft^3}{day})\ (1018.6\ Btu/ft^3\ )} = 0.96$$

By including required conversion factors, Eq. (2) can be modified as

$$E_{vent/HEX} = \rho Q_{vent/HEX}\, c_p\, \Delta T_{rise}\, C_2 C_9 \qquad (E1)$$

Including the modified $E_{vent/HEX}$ of Eq. (E1) in Eq. (4a), and using the required numerical values, η_{plant} is

$$\eta_{plant} = 0.96 + \frac{\left(61\,\frac{lb_m}{ft^3}\right)(87\,gpm)\left(1.004\,\frac{Btu}{lb_m{}^{\circ}\mathrm{F}}\right)(8.33\,^{\circ}\mathrm{F})\left(0.00228\,\frac{ft^3}{s\,gpm}\right)\left(86400\,\frac{s}{day}\right)+\left(61\,\frac{lb_m}{ft^3}\right)(14.48\,gpm)\left(1.004\,\frac{Btu}{lb_m{}^{\circ}\mathrm{F}}\right)(17.06\,^{\circ}\mathrm{F})\left(0.00228\,\frac{ft^3}{s\,gpm}\right)\left(86400\,\frac{s}{day}\right)}{\left(586300\,\frac{ft^3}{day}\right)\left(1018.6\,Btu/ft^3\right)}$$

$$= 0.98$$

Here, 193.3 Btu/lb_m and 1197.7 Btu/lb_m are the enthalpies of boiler feed water (in compressed liquid state) at a temperature of 225 ^{0}F (for a pressure of 175 psig) and saturated steam at 375 ^{0}F (from steam tables). This is explained in Section 3.i. η_{boiler} and η_{plant} were calculated for each day of the month. These calculated efficiencies are tabulated in Table E1. As shown in Table E1, the average boiler efficiency for October was 0.86 while the average boiler efficiency for November was 0.92. The average plant efficiency for October was 0.88 while the average boiler efficiency for November was 0.94.

There was fluctuation in the boiler efficiencies. On close observation, it was found that, from October 2 to October 7 of 2015, only Boiler #7 was used to produce steam. From October 9 to October 31, Boiler #8 was used to generate steam. On October 1 and October 8, both Boilers #7 and #8 were used. Boiler #7 is a relatively new boiler and hence, it can be assumed that the efficiency of Boiler #7 is higher than that of Boiler #8. The average efficiency for Boiler #7 was 0.96 while that for Boiler #8 was 0.84.

The same trends were found for November of 2015. Boiler #8 was used alone from November 1 to November 3 of 2015. The average boiler efficiency was 0.83 for those three days. From November 5 to November 8, November 10 to November 12, and November 14 to November 17, Boiler #7 was used individually, and the average boiler efficiency was higher, ~ 0.95. On November 13 and from November 18 to November 30, Boiler #7 was used along with Boiler #1. Boiler #1 being an old boiler, the overall average boiler efficiency decreased to 0.93.

The boiler and plant efficiencies calculated in this Appendix have been used in Appendix F for the calculation of natural gas savings from use of the vent condenser and the basement heat exchanger.

Table E1: Calculated boiler and plant efficiencies for October and November of 2015

Day in October 2015	Boiler Number	Calculated Boiler Efficiency	Calculated Plant Efficiency	Day in November 2015	Boiler Number	Calculated Boiler Efficiency	Calculated Plant Efficiency
1	7 & 8	0.96	0.98	1	8	0.84	0.86
2	7	0.95	0.97	2	8	0.82	0.84
3	7	0.95	0.97	3	8	0.84	0.86
4	7	0.96	0.98	4	7 & 8	0.90	0.92
5	7	0.96	0.98	5	7	0.95	0.97
6	7	0.96	0.98	6	7	0.95	0.97
7	7	0.96	0.98	7	7	0.95	0.97
8	7 & 8	0.86	0.88	8	7	0.95	0.97
9	8	0.84	0.86	9	2, 7 & 8	0.90	0.92
10	8	0.83	0.85	10	7	0.95	0.97
11	8	0.82	0.84	11	7	0.95	0.97
12	8	0.84	0.86	12	7	0.95	0.97
13	8	0.82	0.84	13	1 & 7	0.93	0.95
14	8	0.82	0.84	14	7	0.95	0.97
15	8	0.83	0.85	15	7	0.96	0.98
16	8	0.85	0.87	16	7	0.93	0.95
17	8	0.84	0.86	17	7	0.94	0.96
18	8	0.83	0.85	18	1 & 7	0.91	0.93
19	8	0.84	0.86	19	1 & 7	0.93	0.95
20	8	0.84	0.86	20	1 & 7	0.93	0.95
21	8	0.84	0.86	21	1 & 7	0.93	0.95
22	8	0.83	0.85	22	1 & 7	0.93	0.95
23	8	0.83	0.85	23	1 & 7	0.93	0.95
24	8	0.84	0.86	24	1 & 7	0.93	0.95
25	8	0.83	0.85	25	1 & 7	0.95	0.97
26	8	0.83	0.85	26	1 & 7	0.93	0.95
27	8	0.83	0.85	27	1 & 7	0.92	0.94
28	8	0.83	0.85	28	1 & 7	0.92	0.94
29	8	0.83	0.85	29	1 & 7	0.92	0.94
30	8	0.83	0.85	30	1 & 7	0.91	0.93
31	8	0.84	0.86				
Average		**0.86**	**0.88**	**Average**		**0.92**	**0.94**
Standard Deviation		**0.052**	**0.052**	**Standard Deviation**		**0.033**	**0.033**

Appendix F: Heat Exchanger Energy Gain Calculations

By using both the vent condenser and the heat exchanger in the basement, less fuel (i.e., natural gas) was used to heat the condensate/boiler feed water. In this section, the volume of fuel saved by using the vent condenser and basement heat exchanger with respect to the volume saved when the vent condenser and basement heat exchanger were not in use, and subsequently the monetary savings, were calculated on a yearly basis. The inputs used for calculation of the energy savings with respect to the energy saved when the vent condenser and basement heat exchanger were not in use, are listed below.

Average gas $LHV_{fuel} = 1018.6\ \frac{Btu}{ft^3}$ [17]

Water density at 160 ^{0}F, $\rho\ =\ 61\ lb_m/ft^3$

Average temperature rise of condensate water, $\Delta T_{rise}\ =\ 8.31$ °F (see Table F1)

Specific heat of water at 160 ^{0}F, $c_p\ = 1.004\ Btu/(lb_m$ °F)

Conversion factor of flow rate from gpm to ft^3/s, $C_2\ =\ 0.00228\ \frac{ft^3}{s\ gpm}$

Boiler efficiency for October of 2015 (Average from Table E1), $\eta_{boiler}\ =\ 0.86$

Plant efficiency for October of 2015 (Average from Table E1), $\eta_{plant}\ =\ 0.88$

Example calculation of the natural gas savings by using the vent condenser with respect to the natural gas savings when the vent condenser was not in use between 12:12 am and 12:25 am on October 2015

In order to calculate the natural gas savings for the vent condenser with respect to the natural gas savings when the vent condenser was not used, the energy gain of the excess condensate water has to be calculated. The mass flow rate of the condensate water has to be calculated first, which requires the density and volume flow rate of water as inputs in Eq. (F1). Once the mass flow rate (in lb_m/hr) is known, Eq. (2) can be used to calculate the energy gain (in Btu/hr) by the excess condensate water, due to the use of the vent condenser. Because of the energy gain, less natural gas is used in the boiler to produce steam. The volume of natural gas (in ft^3/min) that is saved can be calculated by using Eq. (4b). The natural gas savings can then be calculated by multiplying the volume of gas saved by the cost of natural gas per ft^3.

In order to calculate the mass flow rate ($^{lb_m}/_{hr}$) from volume flow rate (gpm), use

$$m_{vent/HEX}\ = \rho\, Q_{vent/HEX}\, C_7 C_8 \qquad (F1)$$

Using Eq. (F1), for a condensate flow rate of 87.24 gpm (see Table F1) to the vent condenser, and C_7 and C_8 being units conversion factors,

$$m_{vent/HEX} = \left(61\,\frac{lb_m}{ft^3}\right)(\,87.25\ gpm\,)\left(0.00228\ \frac{ft^3}{s\ gpm}\right)\left(3600\ \frac{s}{hr}\right) = \ 43{,}685\ {}^{lb_m}/_{hr} \qquad (F2)$$

From Eq. (2), we get,

$$E_{vent/HEX} = \left(43{,}685\ {}^{lb_m}/_{hr}\right)\left(1.004\,\frac{Btu}{lb_m\ °F}\right)(8.31\ °F)$$

$$= 364{,}474\ Btu/hr$$

From Eq. (4b), we get

$$Vol_{fuel,saved} = \frac{364{,}474\ Btu/hr}{(0.88\,)(1018.6\ Btu/ft^3)} = 406.61\ {}^{ft^3}/_{hr}$$

$$= 406.61\ {}^{ft^3}/_{hr}\ x\,\frac{1\ hr}{60\ min} = 6.77\ {}^{ft^3}/_{min}$$

This process was carried out for every minute for the months of October and November of 2015. Table F1 shows the results of calculating the volume of natural gas that was saved during 14 minutes between 12-1 am on October 5 from using the vent condenser.

Table F1. Calculation of volume of natural gas saved by using the vent condenser (for 14 minutes on October 5)

Time (Hr:Min)	Flowrate of condensate water to the vent condenser $Q_{vent/HEX}$ (gpm)	Temperature rise of condensate water ΔT_{rise} (°F)	Mass flow rate of condensate water $m_{vent/Hex}$ (lb_m/hr)	Energy Gain $E_{vent/HEX}$ (Btu/hr)	Corresponding per minute volume of natural gas saved V (ft^3)
12:12 am	87.24	8.33	43683.76	365880	6.80
12:13 am	87.50	8.35	43810.76	367428	6.83
12:14 am	86.86	8.34	43493.26	364591	6.77
12:15 am	87.39	8.34	43757.03	366890	6.82
12:16 am	87.60	8.33	43864.49	367085	6.82
12:17 am	87.18	8.38	43654.45	367609	6.83
12:18 am	87.24	8.38	43683.76	368031	6.84
12:19 am	87.18	8.32	43654.45	365108	6.78
12:20 am	86.92	8.31	43522.57	363524	6.75
12:21 am	87.24	8.27	43683.76	362851	6.74
12:22 am	87.28	8.26	43703.30	362969	6.74
12:23 am	87.24	8.27	43683.76	363027	6.74
12:24 am	87.28	8.25	43703.30	362398	6.73
12:25 am	87.44	8.29	43781.45	364719	6.78
Average	87.25	8.31	43691.44	365150	6.77

Data was logged for every minute in an entire month (October of 2015) from Table F1. The volume of natural gas saved was calculated for each minute for the entire month of October by using Eq. (4b). All of the per-minute volumes were summed to give the total monthly volume of natural gas saved for October of 2015 by using the vent condenser and the basement heat exchanger. Calculations were performed in MS Excel.

Equations (2) and (4b) were also used to calculate the volume of natural gas saved for the basement heat exchanger of the power plant. As explained in Section 3.iii, makeup water flow rate in the basement heat exchanger was not recorded on a per-minute basis because there was no flow meter in the makeup water pipeline. So, the flow rate was estimated, based on the temperature rise of the makeup water across the heat exchanger and on the hand-recorded makeup water volumes. The solenoid valves at the entrance of the condensate storage tanks opened/closed automatically. Makeup water flow rate was not continuous for all 60 minutes of each hour. This was because, for certain time periods, the solenoid valves were closed; so there was zero makeup water flow during those time periods. Table F2 shows the results from calculating the volume of natural gas based on selected readings (i.e., between 12 and 1 am on October 5 of 2015) when makeup water was assumed to be flowing.

For the basement heat exchanger, the weighted per-minute makeup water flow rate was calculated based on temperature ratios, as explained in Section 3.iii. The temperature ratio was taken as $\frac{\Delta T_{rise,max}}{\Delta T_{rise}}$, in this case $\frac{33.40\ ^{\circ}\mathrm{F}}{\Delta T_{rise}}$ (see Table 2). So, from Eq. (2), for the basement heat exchanger at 12:05 am

$$E_{vent/HEX} = (m_{vent/HEX}\ \frac{\Delta T_{rise,max}}{\Delta T_{rise}})\ c_p\ \Delta T_{rise} \qquad (F3)$$

The ΔT_{rise} cancels out in the equation. Because of this, the Eq. (F3) energy gain by the basement heat exchanger was constant for that particular hour, even though the makeup water flow rate and the temperature rise were different. On using the ΔT_{rise} values for 12:05 am and 12:06 am, the $E_{vent/HEX}$ by using Eq. (F3) was found to be approximately 118,220 Btu/hr for both of those time points. This value is the same for all recordings during 12-1 am.

The energy reclaimed by both of the heat exchangers was calculated by using the temperature data recorded by the HOBO data logger. However, temperature data for both heat exchangers from January to September of 2015 was not available because the temperature sensors were installed in October of 2015. So, from January to September of 2015, the reclaimed energy was estimated, based on the ratios of the steam generated for the corresponding month to that of steam generated in October or November of 2015. The monthly steam generated data was available from the power plant's boiler data log sheets (see Figs. E1 and E2). This steam generated ratio was then multiplied by the energy reclaimed in October of 2015 (for warmer weather months, i.e., April to September) or that of November of 2015 (for colder weather months, i.e., December to March).

Table F2. Calculation of volume of natural gas saved by using the basement heat exchanger (for selected readings on October 5)

Time (Hour:Minutes)	Makeup water flow rate $Q_{vent/HEX}$ (gpm)	Temperature rise in makeup water (ΔT_{rise}) (°F)	Mass flow rate of makeup water $\dot{m}$ (lb_m/hr)
12:05 am	13.82	17.10	6761
12:06 am	12.60	18.75	6164
12:13 am	12.62	18.73	6174
12:14 am	15.93	14.84	7794
12:15 am	13.82	17.10	6761
12:16 am	13.43	17.60	6570
12:17 am	12.25	19.29	5993
12:18 am	17.68	13.36	8650
12:19 am	13.76	17.17	6732
12:20 am	12.79	18.48	6257
12:21 am	13.87	17.04	6786
12:22 am	13.76	17.17	6732
12:23 am	13.82	17.10	6761
12:24 am	18.12	13.04	8865
12:25 am	14.24	16.60	6967
12:26 am	14.80	15.97	7241
12:27 am	15.30	15.44	7485
12:28 am	17.43	13.56	8527
12:29 am	18.06	13.08	8836
12:30 am	15.96	14.81	7808
12:31am	16.15	14.63	7901
12:38 am	17.74	13.32	8679
12:39 am	16.72	14.13	8180
12:40 am	12.62	18.73	6174
12:41 am	17.35	13.62	8488
12:42 am	12.25	19.29	5993
12:43 am	8.06	29.30	3943
12:44 am	7.07	33.40	3459
12:45 am	15.56	15.18	7613
12:46 am	16.86	14.02	8249
Average	14.48	17.06	7085

In order to calculate the annual energy reclaimed by the vent condenser or basement heat exchanger, Eqs. (F4) and (F5) were used.

$$E_{vent/HEX,month}\left(Warm\ months\ (April\ to\ September)\right) = E_{vent/HEX,month}\left(October,\ 2015\right)\frac{S_G\left(for\ Desired\ Month\right)}{S_G\left(for\ October,\ 2015\right)} \qquad (F4)$$

$$E_{vent/HEX,month}\left(Cool\ months\ (December\ to\ March)\right) = E_{vent/HEX,month}\ (November, 2015)\frac{S_G\left(for\ Desired\ Month\right)}{S_G\left(for\ November,\ 2015\right)} \qquad (F5)$$

Here $E_{vent/HEX,month}$ is the monthly energy reclaimed by the vent condenser or basement heat exchanger, and S_G is the monthly steam generated in lb_m.

Notice that the number of days in each month cancel out as shown in Eq. (F5.a), using April, 2015 as an example. Therefore, the number of days in a month did not affect the calculations.

$$E_{vent/HEX,month}\ (April, 2015) = E_{vent/HEX,month}\ (October, 2015)\left(\frac{30\ days}{31\ days}\right)\frac{\frac{S_G\ \ (for\ April, 2015)}{30\ days}}{\frac{S_G\ (for\ October, 2015)}{31\ days}} \qquad (F5.a)$$

$$= E_{vent/HEX,month}[(October, 2015)]\frac{S_G\ (for\ April, 2015)}{S_G\left(for\ October,\ 2015\right)}$$

Based on Eqs. (F4) and (F5), the monthly energy consumption was estimated for both the vent condenser and the heat exchanger in the basement and is shown in Table F3.

The total energy reclaimed by the vent condenser and that by the basement heat exchanger in 2015 was approximately 4,787,619,000 Btu and 1,325,550,000 Btu, respectively (see Table F3). The total 2015 volume of natural gas saved by the vent condenser and the basement heat exchanger was approximately 5,222,440 ft^3 and 1,445,940 ft^3, respectively (see Table F4). Based on a cost of \$0.0053 per ft^3 of natural gas [17], the total 2015 savings for natural gas by using the vent condenser was approximately \$27,680 and by using the basement heat exchanger was approximately \$7,660 in 2015. Those savings values were compared to the baseline: the vent condenser or basement heat exchanger was not in use.

The energy reclaimed by the vent condenser was more than double the energy reclaimed by the basement heat exchanger. This is because the average flow rate of the water to the vent condenser was four times higher than the flow rate of the makeup water to the basement heat exchanger, even though temperature rise across the basement heat exchanger was approximately twice than that of the vent condenser.

Table F3. Estimated energy reclaimed by each heat exchanger in each month of 2015

Month of 2015	Steam Generated S_G $(lb_m\ /month)$	Energy reclaimed by vent condenser $E_{vent/HEX,month}$ (Btu/month)	Energy reclaimed by basement heat exchanger $E_{vent/HEX,month}$ (Btu/month)
January	37,821,700	662,064,000 **(estimated)**	183,307,000 **(estimated)**
February	38,179,100	668,321,000 **(estimated)**	185,039,000 **(estimated)**
March	27,418,400	479,956,000 **(estimated)**	132,886,000 **(estimated)**
April	19,858,400	347,619,000 **(estimated)**	96,246,000 **(estimated)**
May	15,015,900	262,851,000 **(estimated)**	72,776,000 **(estimated)**
June	14,563,600	254,933,000 **(estimated)**	70,584,000 **(estimated)**
July	14,423,800	252,486,000 **(estimated)**	69,906,000 **(estimated)**
August	14,359,000	251,352,000 **(estimated)**	69,592,000 **(estimated)**
September	14,286,400	250,081,000 **(estimated)**	69,240,000 **(estimated)**
October	18,118,300	317,157,000 **(computed)**	87,811,000 **(computed)**
November	26,090,300	456,705,000 **(computed)**	126,447,000 **(computed)**
December	33,367,700	584,094,000 **(estimated)**	161,716,000 **(estimated)**
Total (For Calendar Year 2015)	**273,502,600 lbm/yr**	**4,787,619,000 Btu/yr (estimated)**	**1,325,550,000 Btu/yr (estimated)**

Table F4. Estimated volume of natural gas saved by each heat exchanger in 2015

Month of 2015	Volume of Natural Gas Saved by Vent Condenser $Vol_{fuel,saved}$ (ft^3)	Volume of Natural Gas Saved by Basement Heat Exchanger $Vol_{fuel,saved}$ (ft^3)
October	353,824 **(computed)**	97,963 **(computed)**
November	476,984 **(computed)**	132,061 **(computed)**
Total (For Calendar Year 2015)	**5,222,440 (estimated)**	**1,445,940 (estimated)**

Appendix G: Pump Performance Curves and Specifications

This appendix contains the pump performance curves of the Worthington D-824 constant speed pump and the Grundfos CRE 15-3 variable speed pumps. These pump performance curves were used to obtain the pump power consumption and the pump efficiency, when the discharge flow rate was known.

G1. Worthington D-824 Constant Speed Pump Curves and Specifications

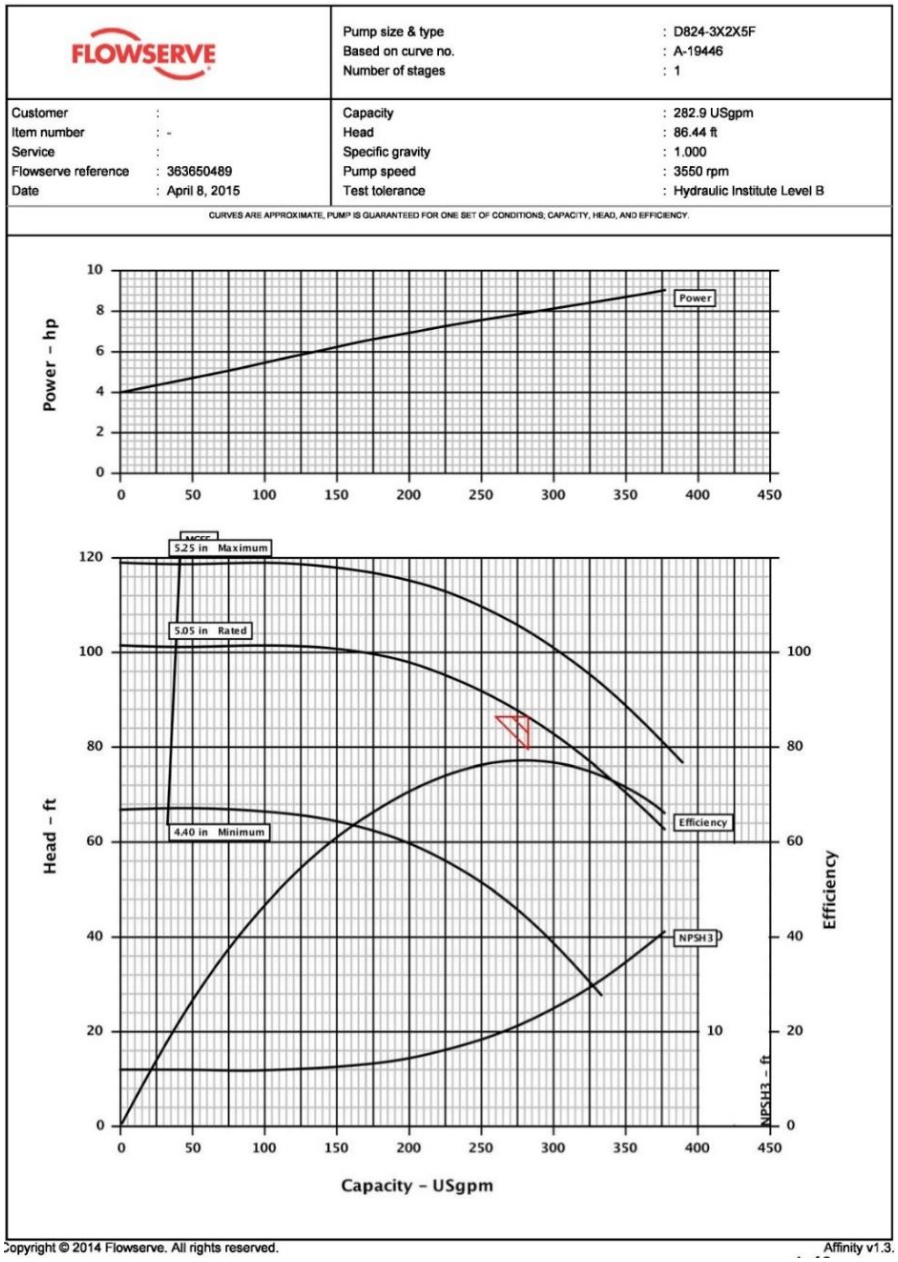


Fig. G1a. Performance curves for Worthington D-824 constant speed pump (reproduced from Ref. 7)

Knowing the flow rate and the impeller diameter of the pump (i.e., 5.05 inches), the pump's power consumption and the efficiency were read from Fig. G1a. For example, when a flow rate of 150 gpm was considered, the corresponding power consumption of the pump was approximately 6.4 hp, and the pump efficiency was 0.6. This was done using pencil and straight edge, since no online version of this curve was available. There is no information available about whether the efficiency read from this curve is the overall pump efficiency or just the motor efficiency. For this thesis, the efficiency read from this pump curve was considered to be overall pump efficiency. In Fig. G1a, the pump performance curve for the Worthington D-824 pump is shown, while Figs. G1b and G1c show the technical and construction specifications of the Worthington pump.

FLOWSERVE

Hydraulic Datasheet

Customer	:	Pump / Stages	: D824-3X2X5F / 1
Customer reference	: Douglas Pump	Based on curve no.	: A-19446
Item number	: -	Flowserve reference	: 363650489
Service	:	Date	: April 8, 2015

Operating Conditions

Capacity	: 282.9 USgpm
Water capacity (CQ=1.00)	: -
Normal capacity	: -
Total developed head	: 86.44 ft
Water head (CH=1.00)	: -
NPSH available (NPSHa)	: Ample
NPSHa less NPSH margin	: -
Maximum suction pressure	: 0.0 psig

Liquid

Liquid type	: Other	
Temperature / Spec. Gravity	: 60 F	/ 1.000
Solid Size - Actual / Limit	: -	/ 0.50 in
Viscosity / Vapor pressure	: 1.00 cSt	/ -

Materials / Specification

Material column code	: STD
Pump specification	: General Industrial

Other Requirements

Hydraulic selection : No specification
Construction : No specification
Test tolerance : Hydraulic Institute Level B
Driver Sizing : Max Power(MCSF to EOC)with SF

Performance

Hydraulic power	: 6.18 hp
Pump speed	: 3550 rpm
Pump overall efficiency (CE=1.00)	: 77.5 %
NPSH required (NPSH3)	: 11.1 ft
Rated power	: 7.97 hp
Maximum power	: 9.04 hp
Driver power	: 10.00 hp / 7.46 kW
Casing working pressure (based on shut off @ cut dia/rated SG)	: 43.9 psig
Maximum allowable	: 175.0 psig
Hydrostatic test pressure	: 265.0 psig
Est. rated seal chamb. press.	: -

Impeller diameter	
Rated	: 5.05 in
Maximum	: 5.25 in
Minimum	: 4.40 in
Suction specific speed	: 9190 US units
Minimum continuous flow	: 38.5 USgpm
Maximum head @ rated dia	: 101.36 ft
Flow at BEP	: 282.9 USgpm
Flow as % of BEP	: 100.0 %
Efficiency at normal flow	: -
Impeller dia ratio (rated/max)	: 96.2 %
Head rise to shut off	: 17.3 %
Total head ratio (rated/max)	: 82.6 %

CURVES ARE APPROXIMATE, PUMP IS GUARANTEED FOR ONE SET OF CONDITIONS; CAPACITY, HEAD, AND EFFICIENCY.

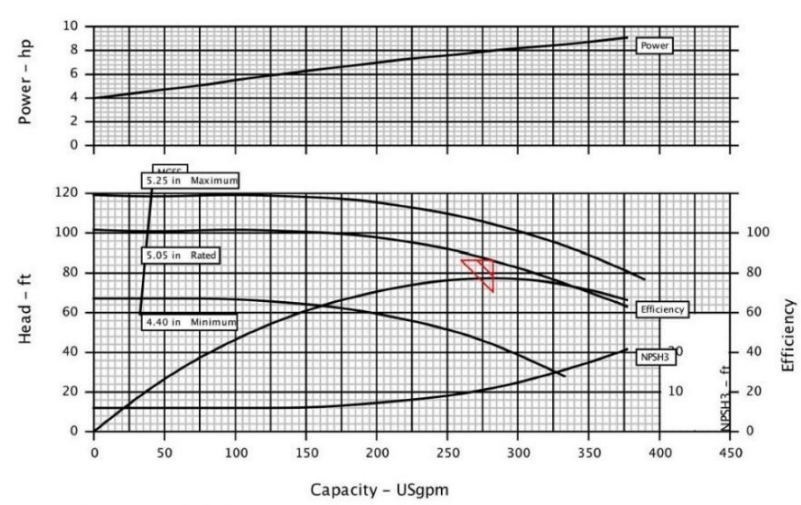


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Fig. G1b. Technical specifications of Worthington D-824 constant speed pump (reproduced from Ref. 7)

FLOWSERVE

Construction Datasheet

Customer	:	Pump / Stages	: D824-3X2X5F / 1
Customer reference	: Douglas Pump	Based on curve no.	: A-19446
Item number	: -	Flowserve reference	: 363650489
Service	:	Date	: April 8, 2015

Construction

Nozzles	Size	Rating	Face	Pos'n
Suction	3.00	125#	FF	End
Discharge	2.00	125#	FF	Top

Casing mounting	: Foot
Casing split	: Radial
Impeller type	: Closed Impeller
Bearing type (radial)	: N/A
Bearing number (radial)	: N/A
Bearing type (thrust)	: N/A
Bearing number (thrust)	: N/A
Bearing lubrication	: Other
Rotation (view from cplg)	: CW per Hyd. Institute

Materials

Casing	: Cast Iron 25
Impeller	: Bronze
Case wear ring	: N/A
Impeller wear ring	: N/A
Inducer	: N/A
Shaft	: Steel
Sleeve	: Bronze

Baseplate, Coupling and Guard

Baseplate type	: N/A
Baseplate material	: N/A
Coupling manufacturer	: N/A
Coupling size	:
Coupling / Shaft guard	: N/A

Weights (Approx.)

Bareshaft pump(nett)	: 58.0 lb
Baseplate(nett)	: -
Driver(nett)	: 127.0 lb
Shipping gross weight/vol.	: 212.7 lb / 6199 cu.in

Testing

Hydrostatic test	: None
Performance test	: None
NPSH test	: None

Paint and Package

Pump paint	: FPD Standard
Base grout surface prep	: FPD Std.
Shipment type	: Domestic

Driver Information

Manufacturer	: Flowserve Choice
Power	: 10.00 hp / 7.46 kW
Service factor (req'st / act)	: 1.0 / -
Speed	: 3600
Orientation / Mounting	: Horizontal / Foot and flange
Driver Type	: NEMA
Frame-size / material	: 215JM / Aluminum
Enclosure	: TEFC
Hazardous area class	: -
Explosion 'T' rating	: -
Volts / Phase / Hz	: 230/460 / 3 / 60 Hz
Amps-full load/locked rotor	: 12.00 A / 72.10 A
Motor starting	: Direct on line (DOL)
Insulation	: F
Temperature rise	: 80 C
Bearings	: Ball
Lubrication	: Grease
Motor mounted by	: Flowserve

Sound Pressure (dBA @ 1.0 m)

Driver, expected	: -
Pump & driver, estimated	: -

Seal Information

Arrangement	: Sgl. Int. O-Ring
Size	: 1.375
Manufacturer / Type	: Flowserve / PAC 51
Material code (Man'f/API)	: BCFXF / -
Internal neck bushing	: N/A

Gland

Gland material	: N/A
Flush	: N/A
Vent	: N/A
Drain	: N/A
Auxiliary seal device	: N/A

Piping

Seal flush plan	: None
Seal flush construction	: -
Seal flush material	: -
Aux seal flush plan	: None
Aux seal flush construction	: -
Aux seal flush material	: -

Notes

-
-
-
-
Bronze Adapter Wear Ring
-

Affinity v1.3.5

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Fig. G1c. Construction data sheet of Worthington D-824 constant speed pump (reproduced from Ref. 7)

G2. Grundfos CRE 15-3 Variable Speed Pump Curves and Specifications

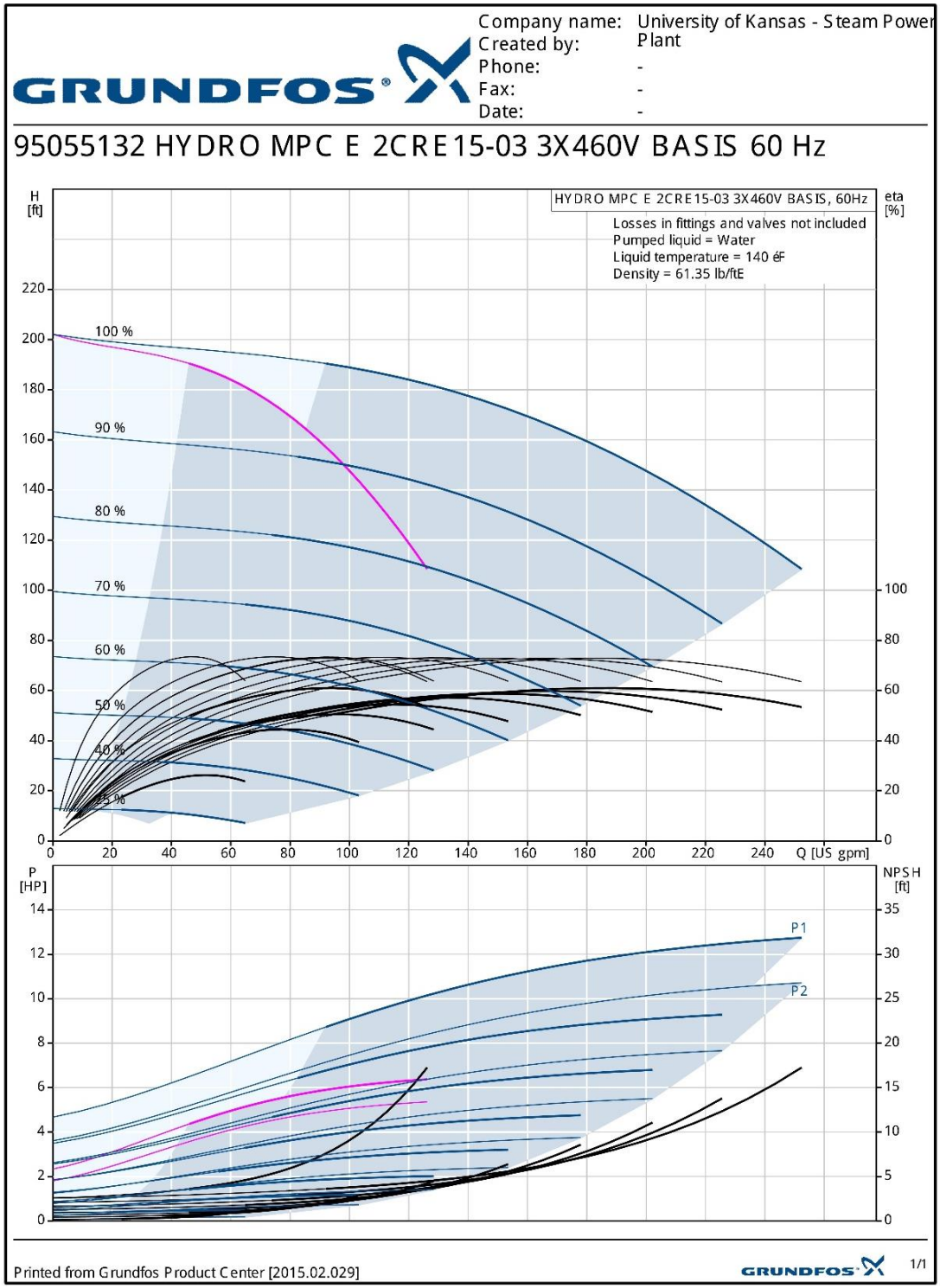
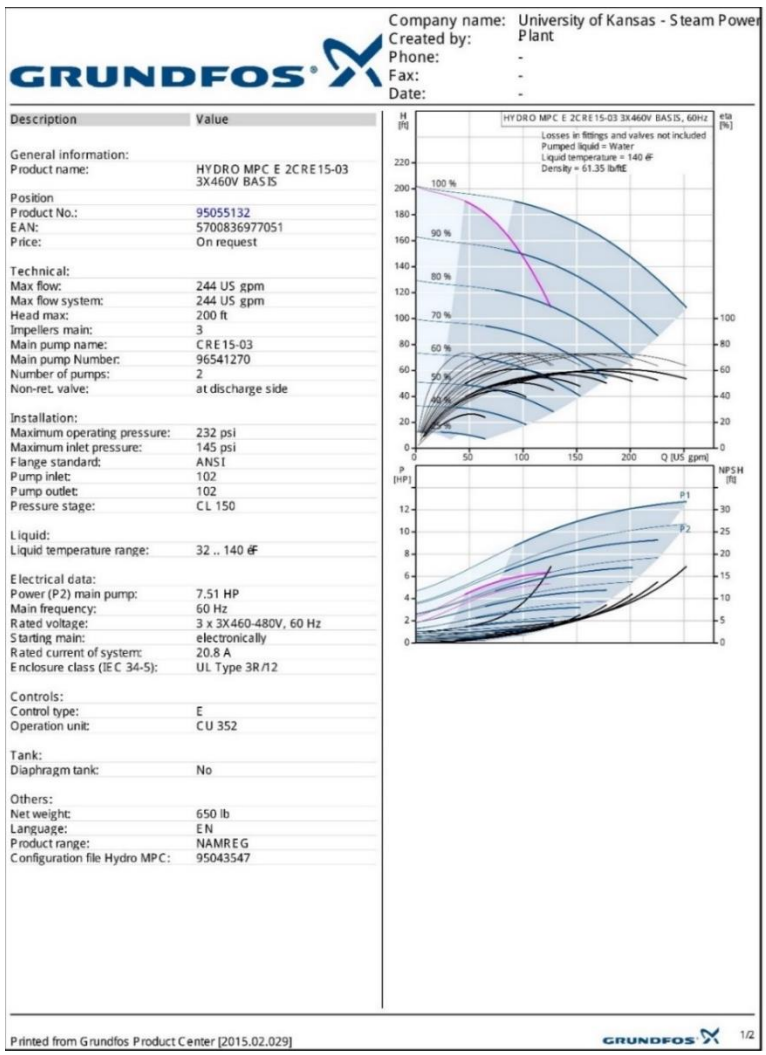


Fig. G2a. Performance curves for Grundfos CRE 15-3 variable speed pump (reproduced from Ref. 37)

Just as in the case for the Worthington pump, knowing the flow rate and the discharge pressure of the Grundfos pumps, the pumps' power consumption and the efficiency were read from Fig. G2a. For example, when a flow rate of 120 gpm was considered along with a discharge head of 50 psi, then the corresponding power consumption of the pumps was approximately 4.584 kW, and the pump efficiency was 0.569. In Fig. G2a, there are two red lines, one in the lower plot and one in the upper plot. The red line in the lower plot indicated the power that the pump delivered to the water (the hydraulic power) while the other indicated the number of pumps in operation. If the input data point was below or on this red line, this meant that only one pump was running. If the input data point was above the red line, this indicated that both pumps were running simultaneously. The curve is available online, and the discharge pressure and the discharge flowrate can be input to give the pump power and efficiency automatically. It is clearly specified in the pump data sheet that the efficiency generated is the overall combined pump, motor and VFD efficiency. This efficiency for the Grundfos pumps was used for this thesis. In Fig. G2a, the performance curve for the Grundfos CRE 15-3 pump is shown, while Fig. H2b shows the technical specifications.



GRUNDFOS

Company name: University of Kansas - Steam Power Plant
Created by:
Phone: -
Fax: -
Date: -

Description	Value
General information:	
Product name:	HYDRO MPC E 2CRE15-03 3X460V BASIS
Position	
Product No.:	95055132
EAN:	5700836977051
Price:	On request
Technical:	
Max flow:	244 US gpm
Max flow system:	244 US gpm
Head max:	200 ft
Impellers main:	3
Main pump name:	CRE15-03
Main pump Number:	96541270
Number of pumps:	2
Non-ret. valve:	at discharge side
Installation:	
Maximum operating pressure:	232 psi
Maximum inlet pressure:	145 psi
Flange standard:	ANSI
Pump inlet:	102
Pump outlet:	102
Pressure stage:	CL 150
Liquid:	
Liquid temperature range:	32 .. 140 °F
Electrical data:	
Power (P2) main pump:	7.51 HP
Main frequency:	60 Hz
Rated voltage:	3 x 3X460-480V, 60 Hz
Starting main:	electronically
Rated current of system:	20.8 A
Enclosure class (IEC 34-5):	UL Type 3R/12
Controls:	
Control type:	E
Operation unit:	CU 352
Tank:	
Diaphragm tank:	No
Others:	
Net weight:	650 lb
Language:	EN
Product range:	NAMREG
Configuration file Hydro MPC:	95043547

Printed from Grundfos Product Center [2015.02.029]

GRUNDFOS 1/2

Fig. G2b. Technical specifications for Grundfos CRE 15-3 variable speed pump (reproduced from Ref. 37)

Appendix H: Data from Pumps

This appendix shows the plots of the flow rate and the power consumption of the Worthington and the Grundfos pumps from June of 2015 to November of 2015. For the data shown in this appendix, the Grundfos pumps were always run in pressure control mode.

H1. Flow Rate of Condensate Water from June of 2015 to November of 2015

The plots for the flow rates of Worthington Pump and the Grundfos Pumps from June of 2015 to November of 2015 are listed in Table H1.

Table H1. List of plots of pump discharge flow rate from June of 2015 to November of 2015

Fig. No.	Time period of recorded data (2015)	Pump
H1	June 1- June 7	Worthington
H2	June 8 - June 15	Grundfos
H3	June 29 – July 6	Worthington
H4	July 13 – July 20	Grundfos
H5	July 28 – August 3	Worthington
H6	August 3 – August 10	Grundfos
H7	August 24 – August 31	Worthington
H8	August 31 – September 7	Grundfos
H9	September 21 – September 28	Worthington
H10	September 28 – October 5	Grundfos
H11	October 19 – October 26	Worthington
H12	November 16 – November 23	Grundfos
H13	November 23 – November 30	Worthington

There are four condensate pumps in the basement of the power plant. The power plant staff switch the pumps every week, mainly for longer lives of the pumps. So, the Worthington and the Grundfos pumps are run only once every month. This explains why there are multi-week skips in the plots listed in Table H1.

For both the Worthington and the Grundfos pumps, the Siemens MagFlow meter was used to record flow rate data. In addition, the Grundfos pumps' control panel also calculated the flow rate of the Grundfos pumps by using the discharge pressure and power consumed by the pumps. Therefore, in the flow rate plots for the Grundfos pumps, there are two separate readings – one from the Siemens recordings and the other from the Grundfos control panel. For the calculations of power consumption for the Grundfos pumps, the flow rate readings from the Siemens flow meter were used. Also, for certain days during the weeks when the Grundfos pumps were running, there was data missing due to the laptop being accidentally disconnected at times.

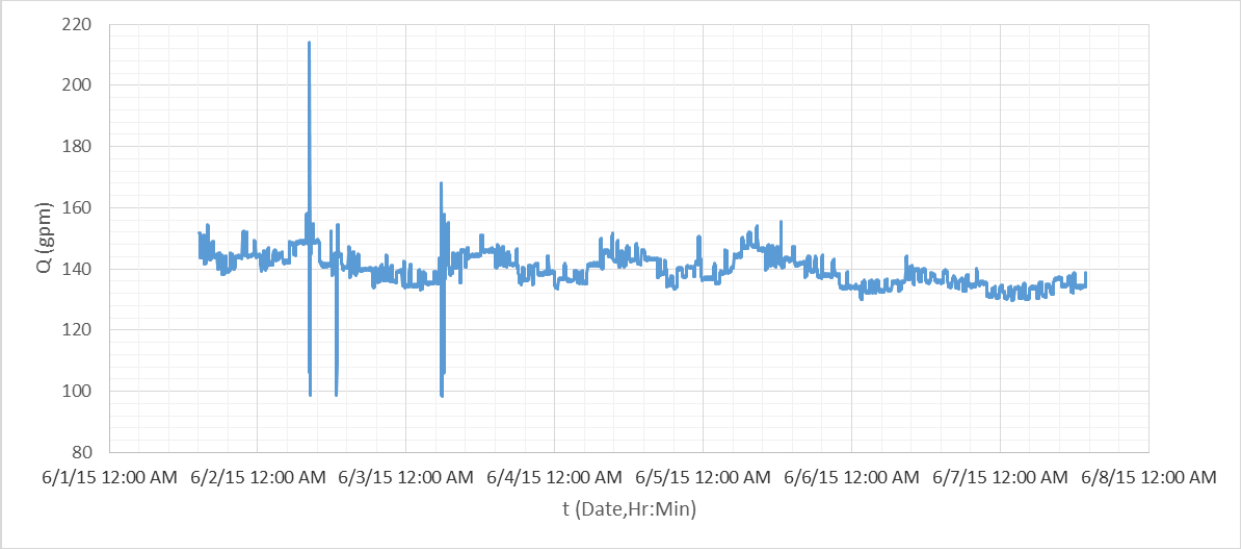


Fig. H1. Flow rate of condensate water when Worthington pump ran from June 1 to June 7 of 2015

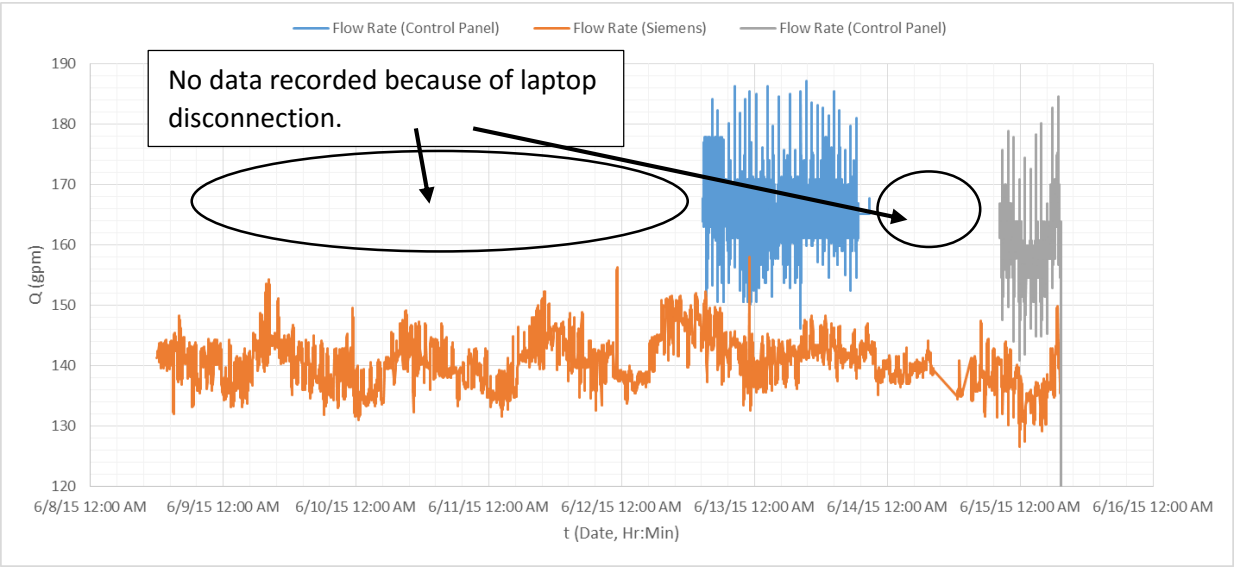


Fig. H2. Flow rate of condensate water when Grundfos pumps ran from June 8 to June 15 of 2015

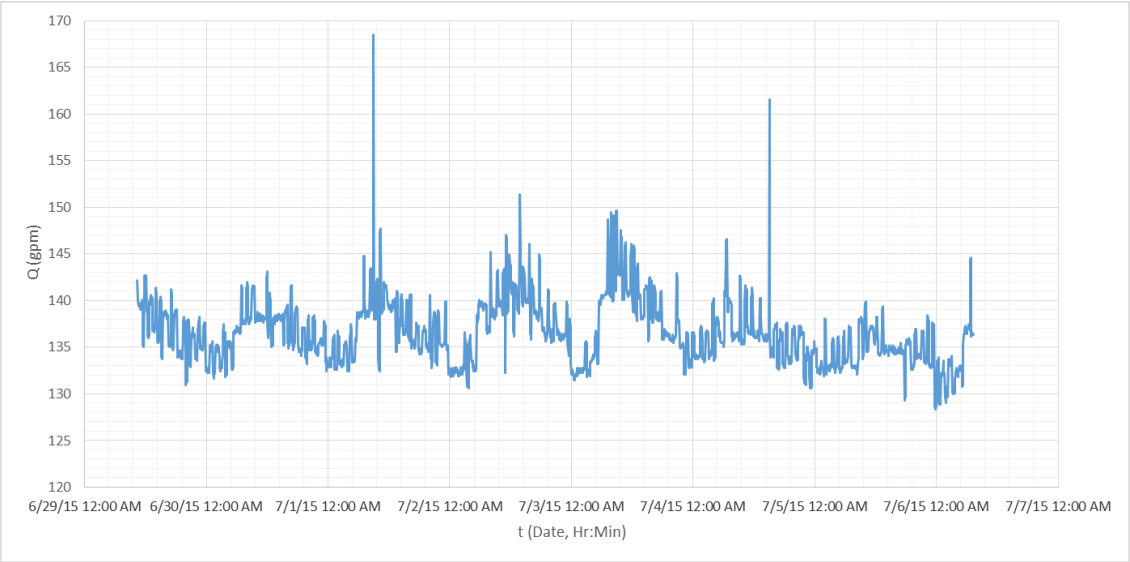


Fig. H3. Flow rate of condensate water when Worthington pump ran from June 29 to July 6 of 2015

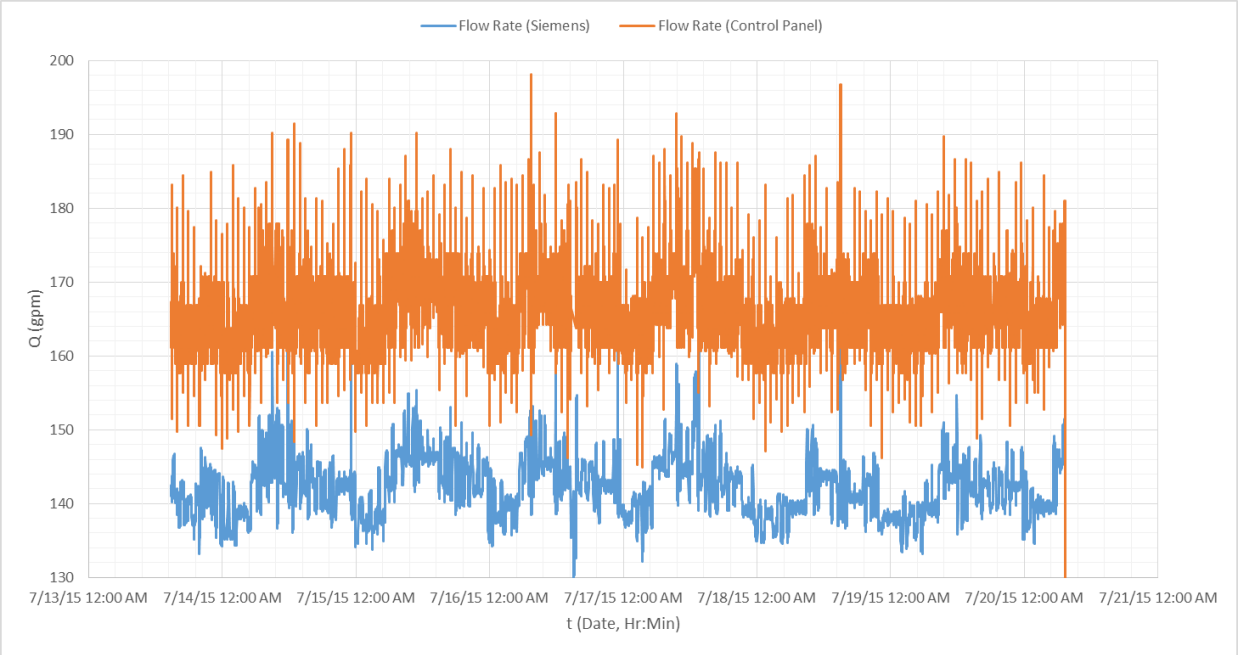


Fig. H4. Flow rate of condensate water when Grundfos pumps ran from July 13 to July 20 of 2015

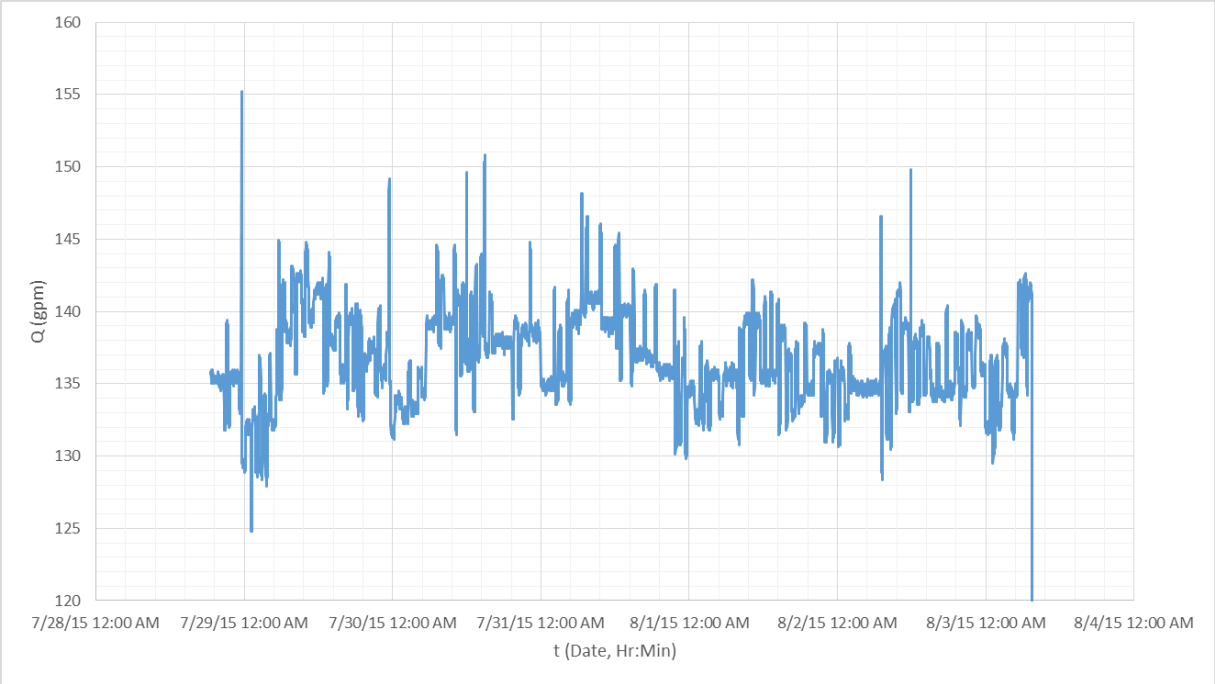


Fig. H5. Flow rate of condensate water when Worthington pump ran from July 28 to August 3 of 2015

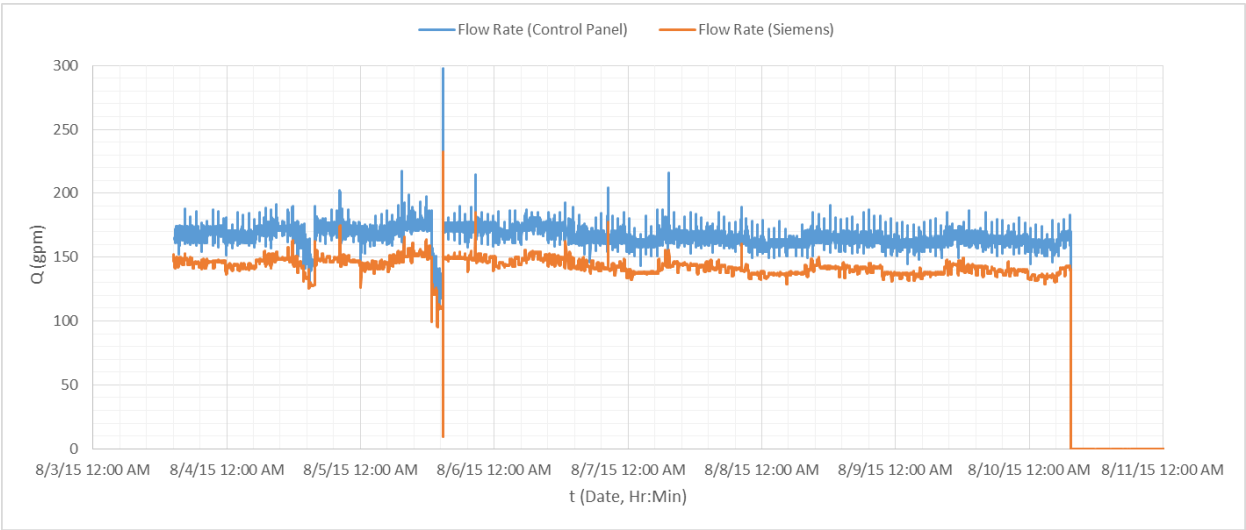


Fig. H6. Flow rate of condensate water when Grundfos pumps ran from August 3 to August 10 of 2015

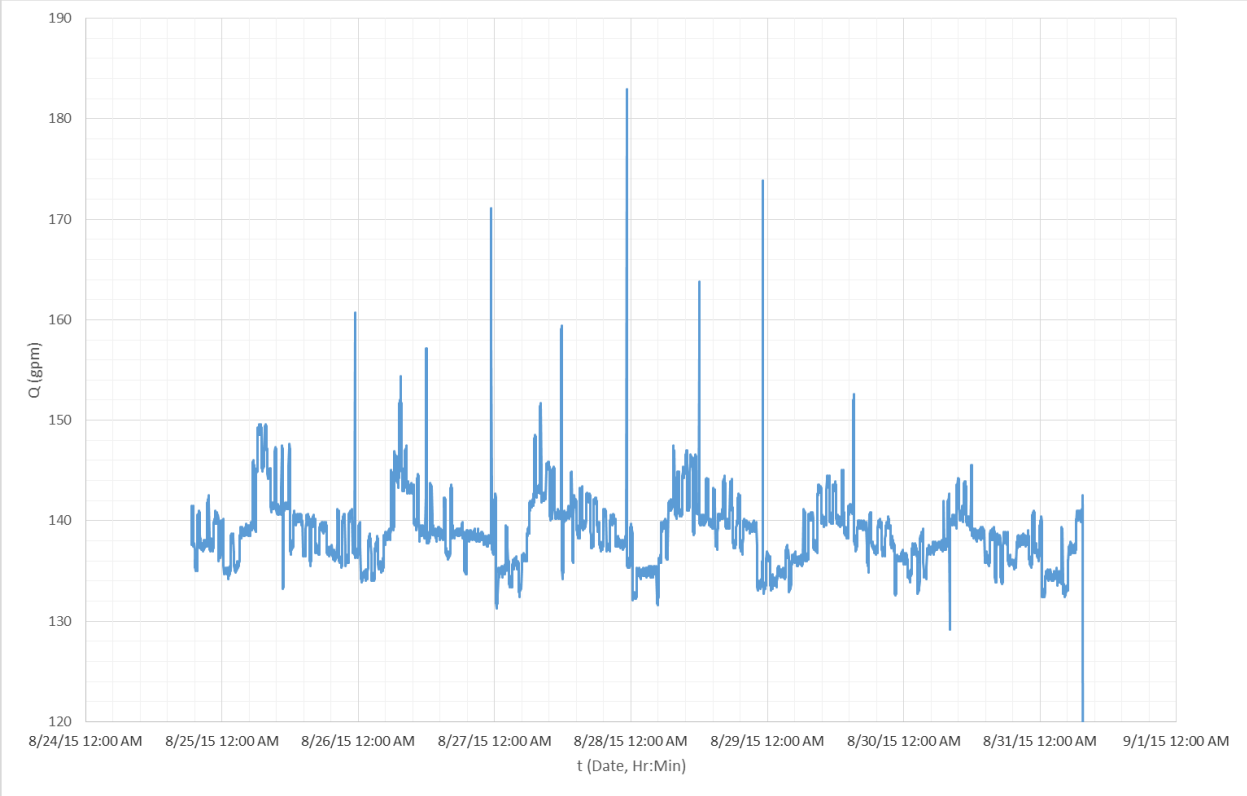


Fig. H7. Flow rate of condensate water when Worthington pump ran from August 24 to August 31 of 2015

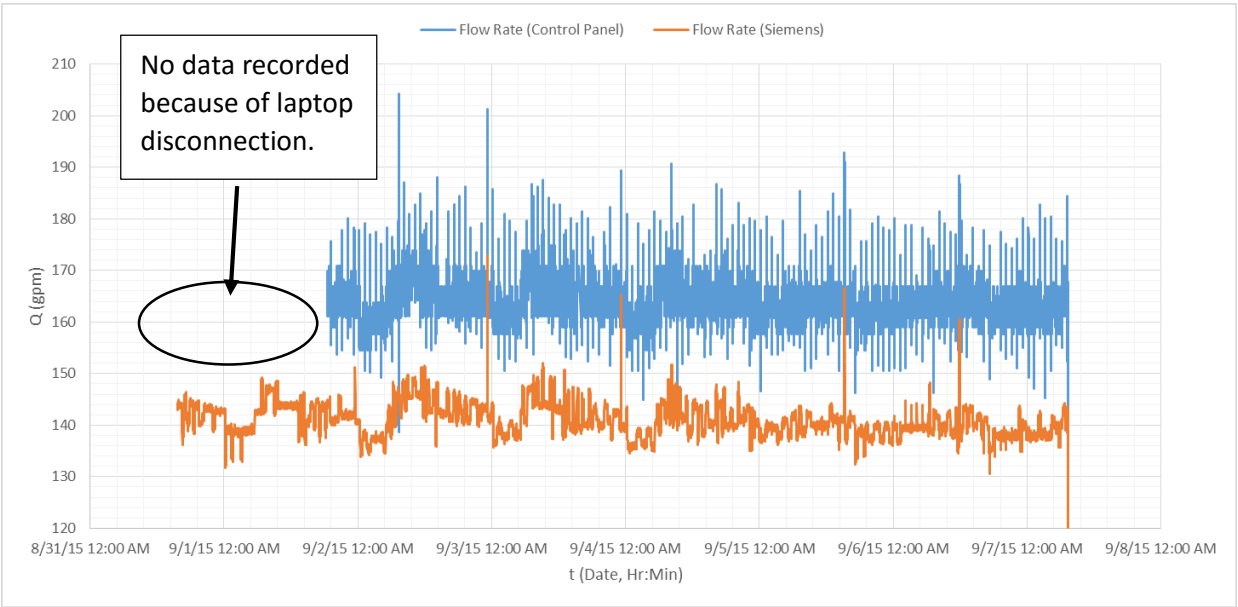


Fig. H8. Flow rate of condensate water when Grundfos pumps ran from August 31 to September 7 of 2015

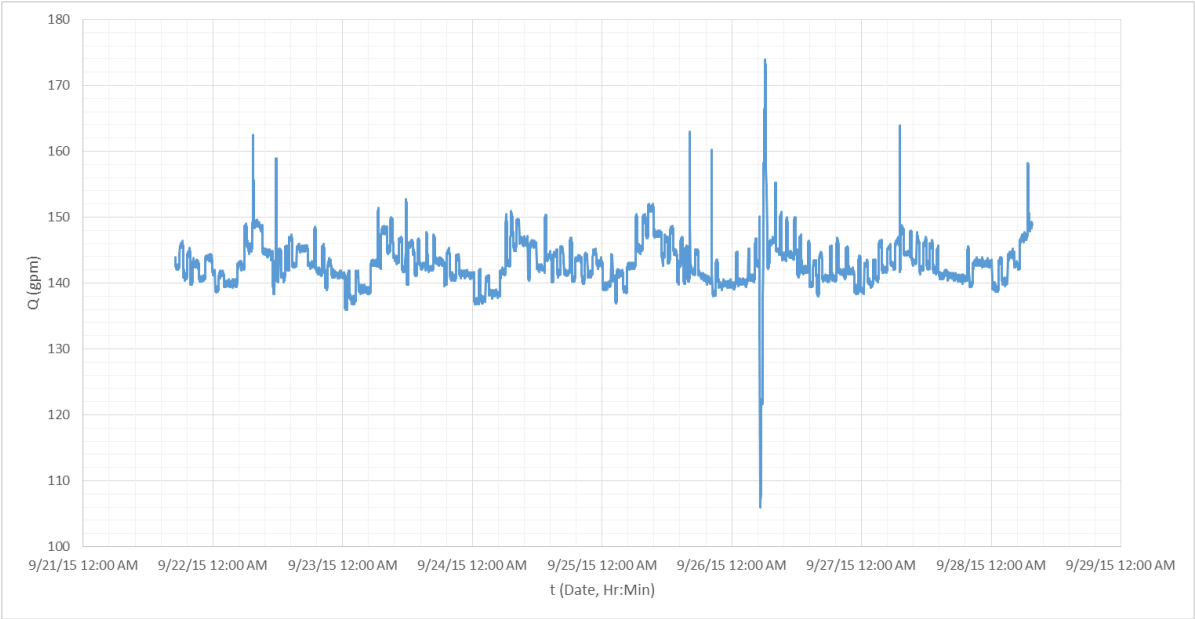


Fig. H9. Flow rate of condensate water when Worthington pump ran from September 21 to September 28 of 2015

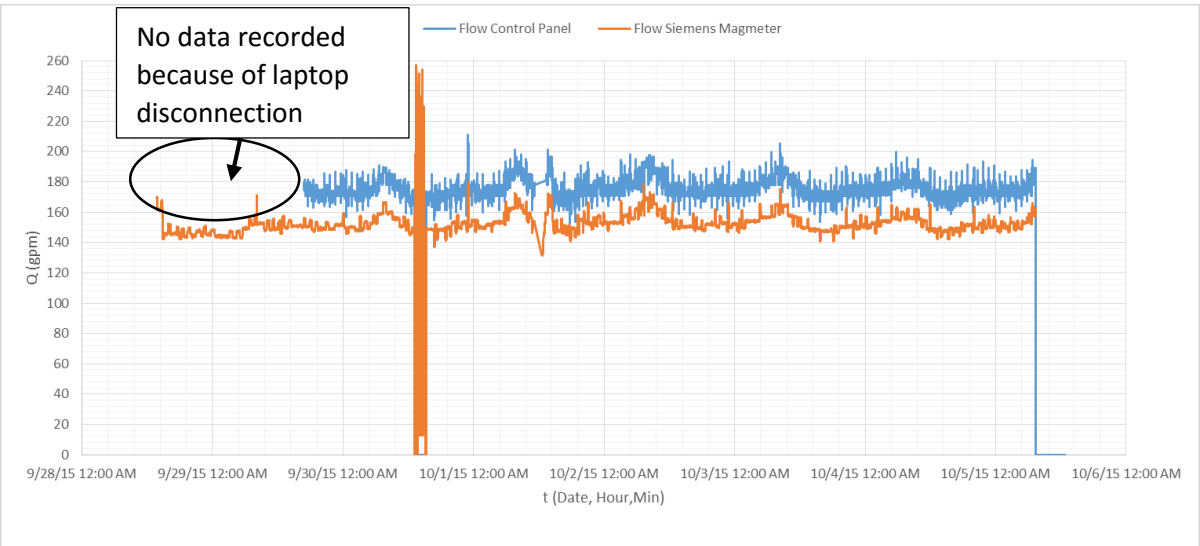


Fig. H10. Flow rate of condensate water when Grundfos pumps ran from September 28 to October 5 of 2015

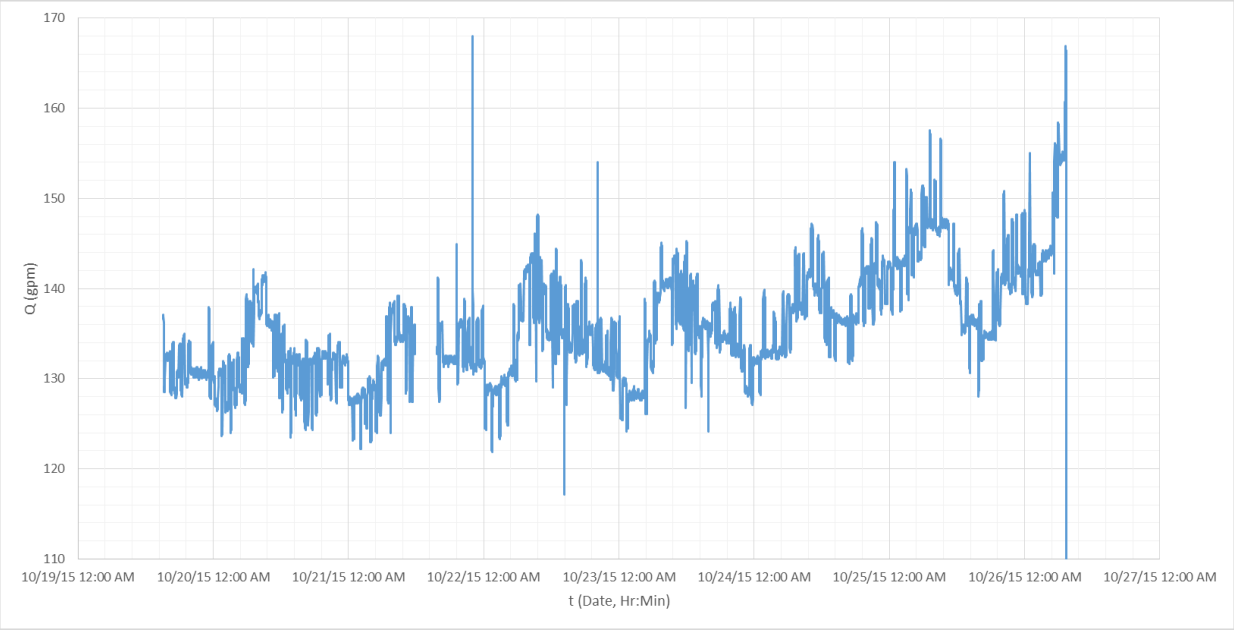


Fig. H11. Flow rate of condensate water when Worthington pump ran from October 19 to October 26 of 2015

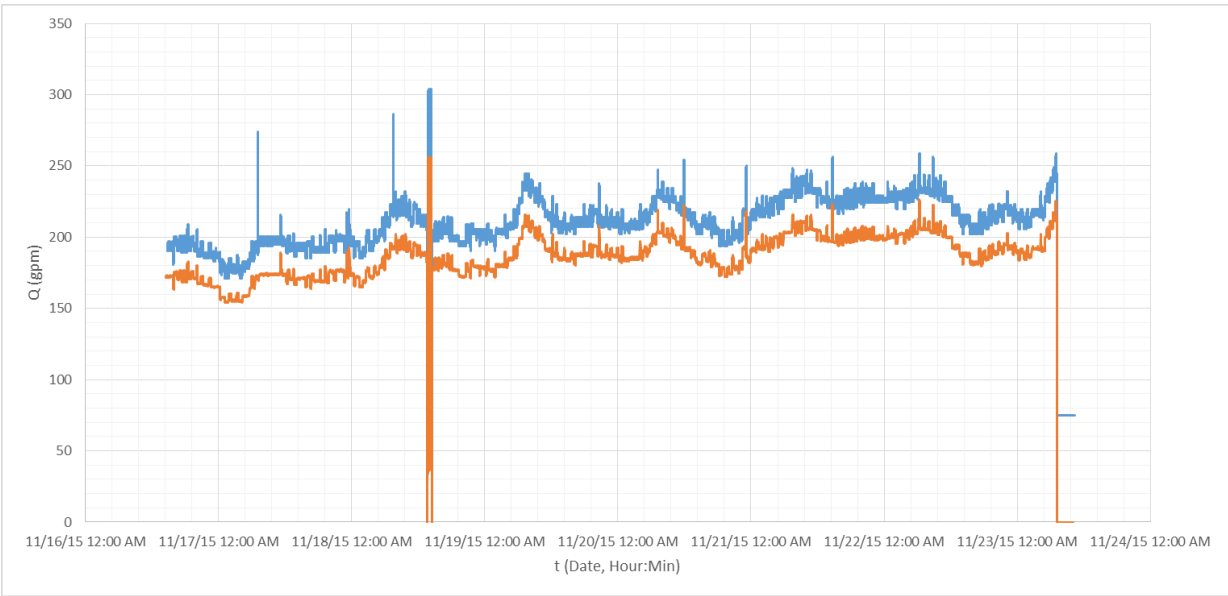


Fig. H12. Flow rate of condensate water when Grundfos pumps ran from November 16 to November 23 of 2015

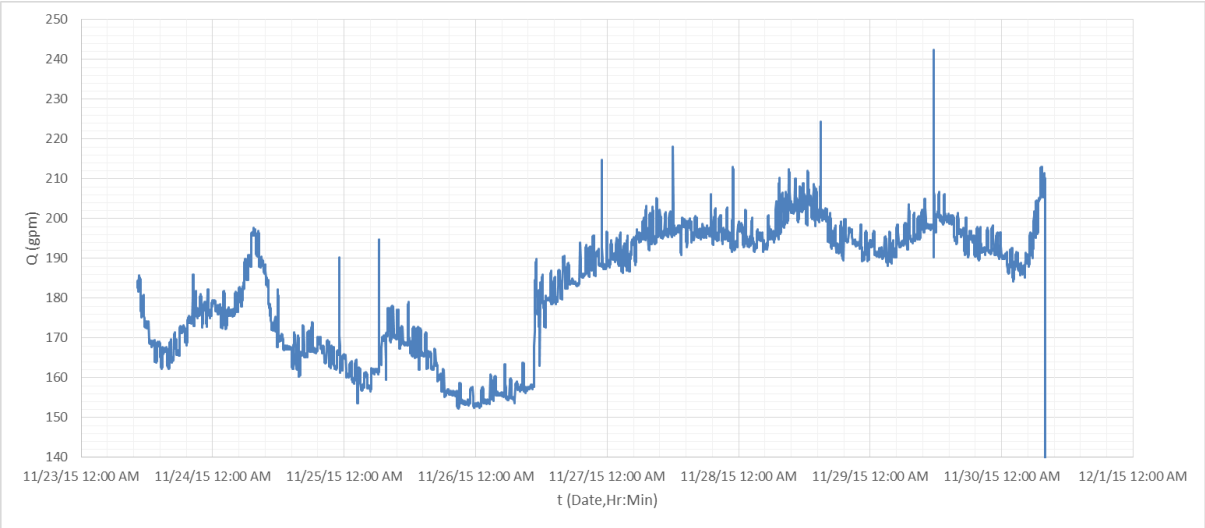


Fig. H13. Flow rate of condensate water when Worthington pump ran from November 23 to November 30 of 2015

H2. Power Consumption of Pumps from June of 2015 to November of 2015

The plots of power consumption of the Worthington Pump and the Grundfos Pumps from June of 2015 to November of 2015 are listed in the Table H2.

Table H2. List of plots for power consumed by pumps from June of 2015 to November of 2015

Fig. No.	Time period of recorded data (2015)	Pump
H14	June 1- June 7	Worthington
H15	June 12 - June 15	Grundfos
H16	June 29 – July 6	Worthington
H17	July 13 – July 20	Grundfos
H18	July 28 – August 3	Worthington
H19	August 3 – August 10	Grundfos
H20	August 24 – August 31	Worthington
H21	September 1 – September 7	Grundfos
H22	September 21 – September 28	Worthington
H23	September 29 – October 5	Grundfos
H24	October 19 – October 26	Worthington
H25	November 16 – November 23	Grundfos
H26	November 23 – November 30	Worthington

The plots for the power consumption for the weeks of June 8 to June 15, and September 28 to October 5, are incomplete in this section. The Grundfos pumps ran during those two weeks. There were connection problems with the Gateway laptop to the Grundfos control panel, due to which, the power consumption of the Grundfos pumps was not recorded for every single day for those two weeks. So, data was taken from June 12 to June 15; and from September 29 to October 5. The power consumption was only recorded for 4 days from August 31 to September 7; while power consumption between 1:30 pm and 3:30 pm on September 30, 2015 was erroneous because level control mode for the Grundfos pumps was run with incorrect settings for the control panel.

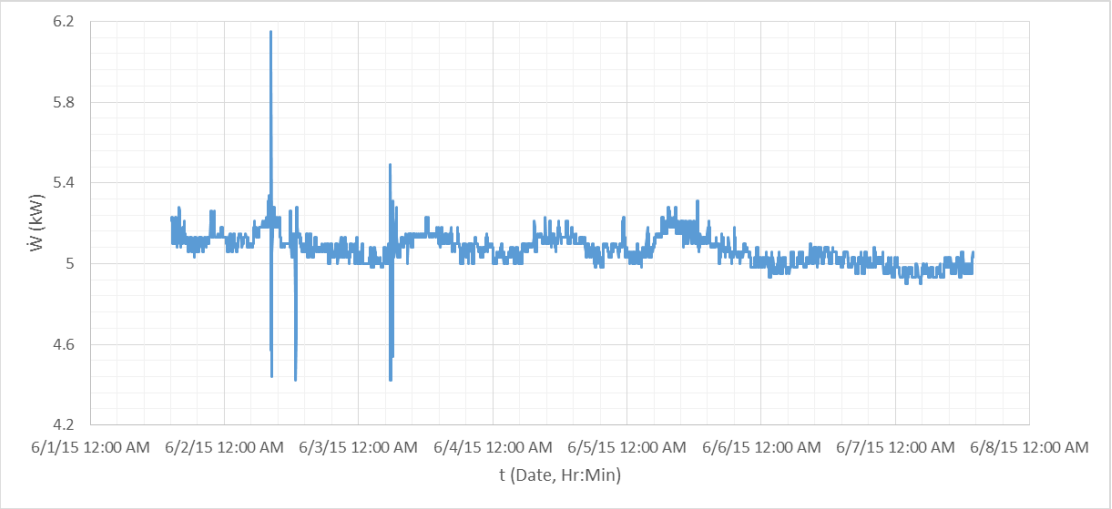


Fig. H14. Power consumption of Worthington pump when it ran from June 1 to June 7 of 2015

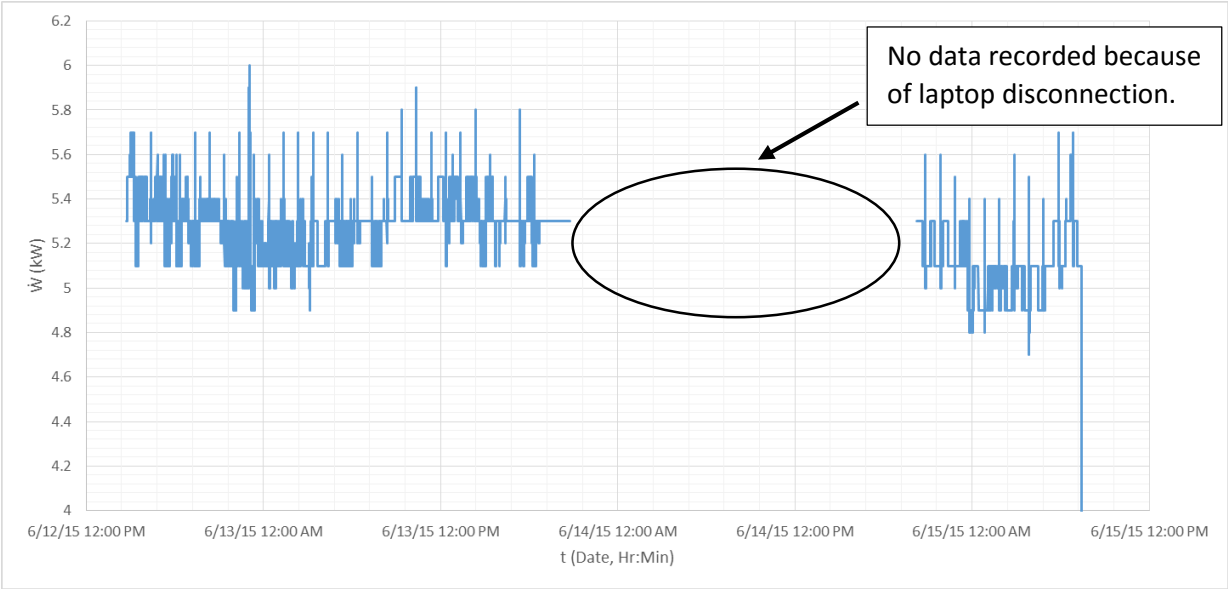


Fig. H15. Power consumption of Grunfos pumps when they ran from June 12 to June 15 of 2015

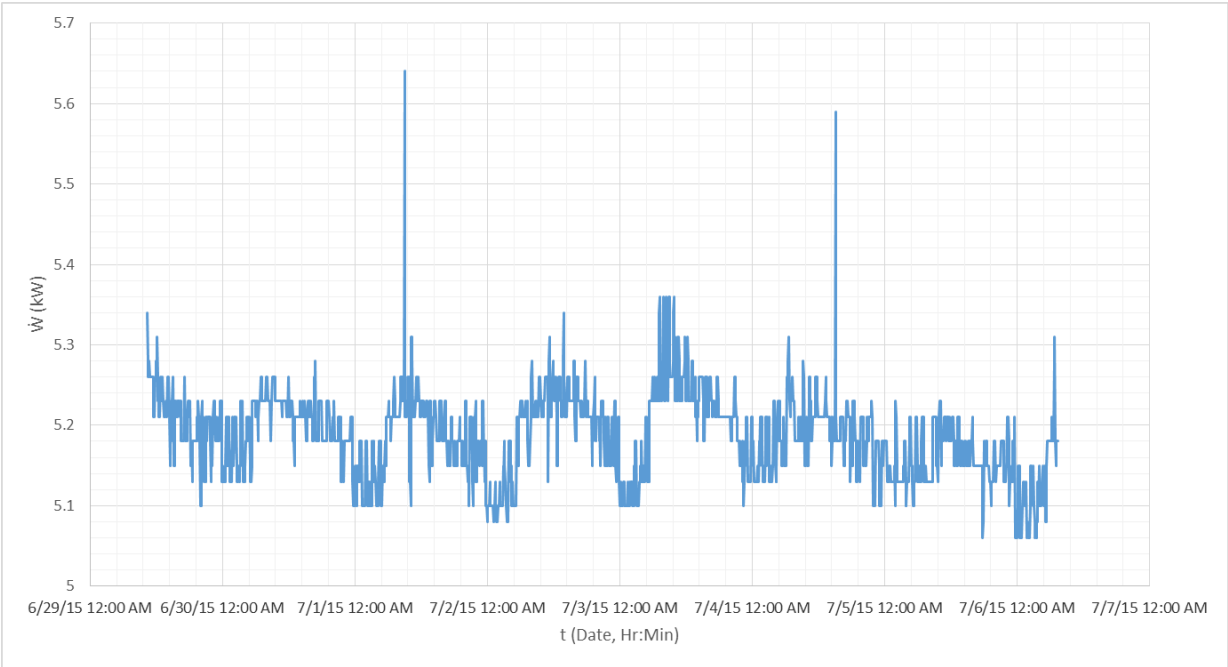


Fig. H16. Power consumption of Worthington pump when it ran from June 29 to July 6 of 2015

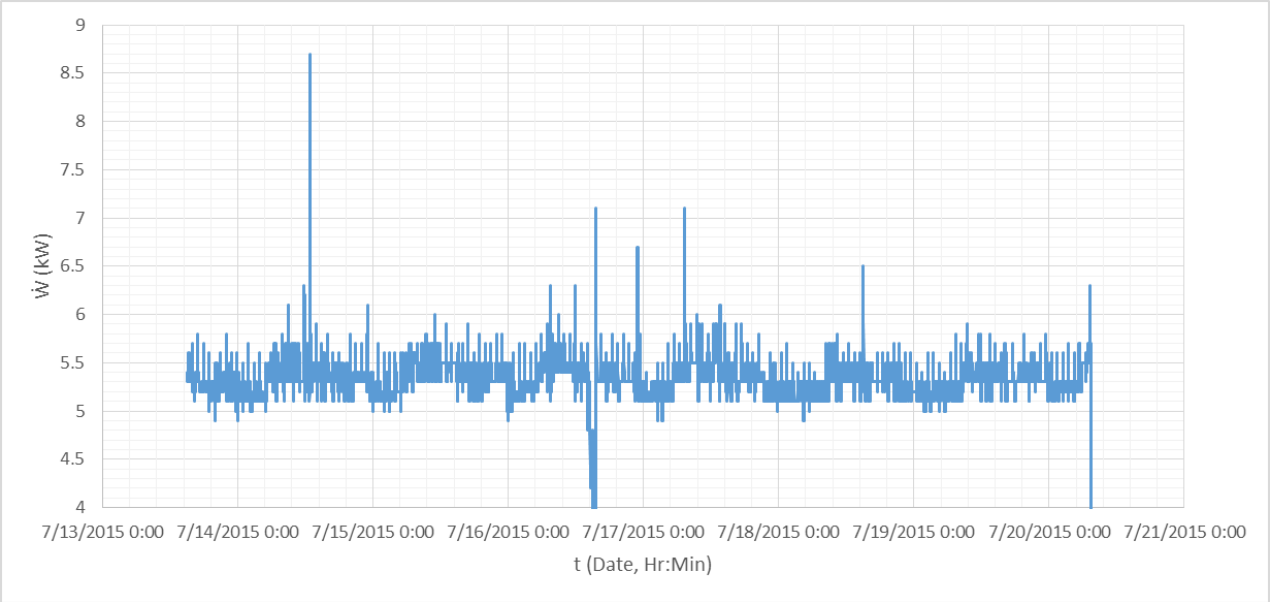


Fig H17. Power consumption of Grundfos pumps when they ran from July 13 to July 20 of 2015

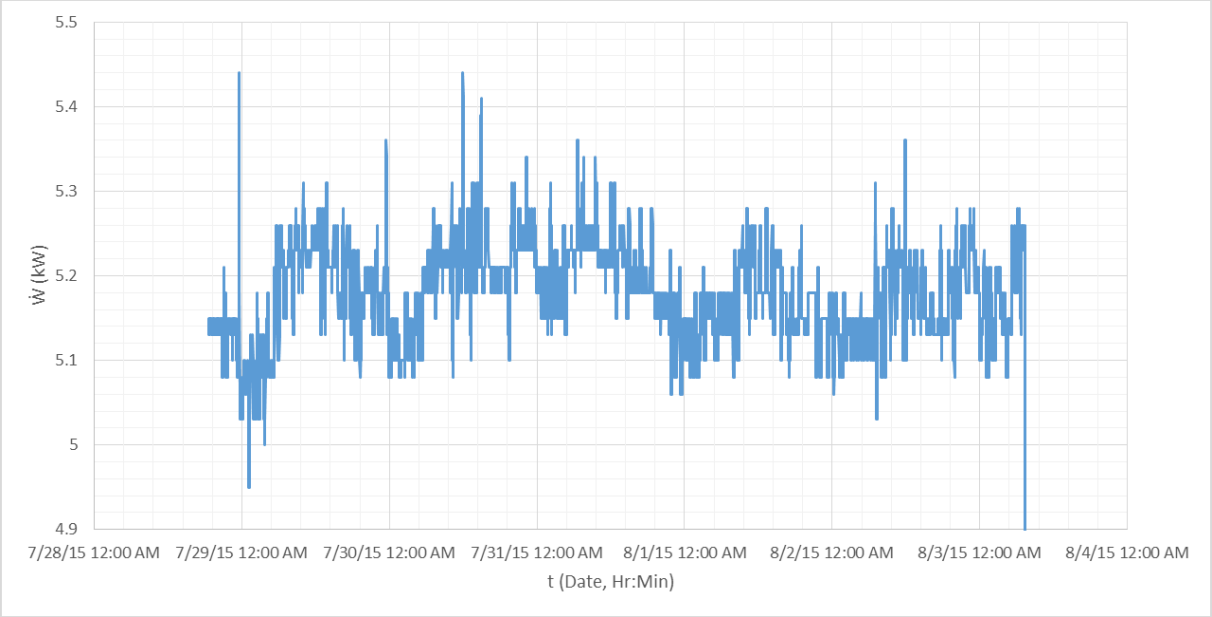


Fig. H18. Power consumption of Worthington pump when it ran from July 28 to August 3 of 2015

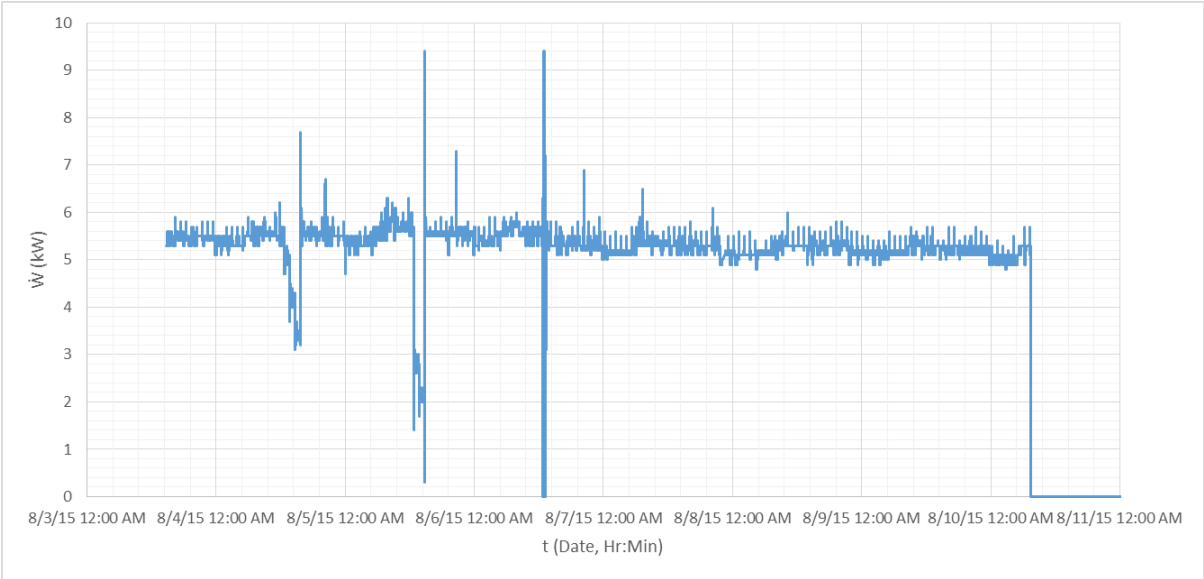


Fig H19. Power consumption of Grundfos pumps when they ran from August 3 to August 10 of 2015

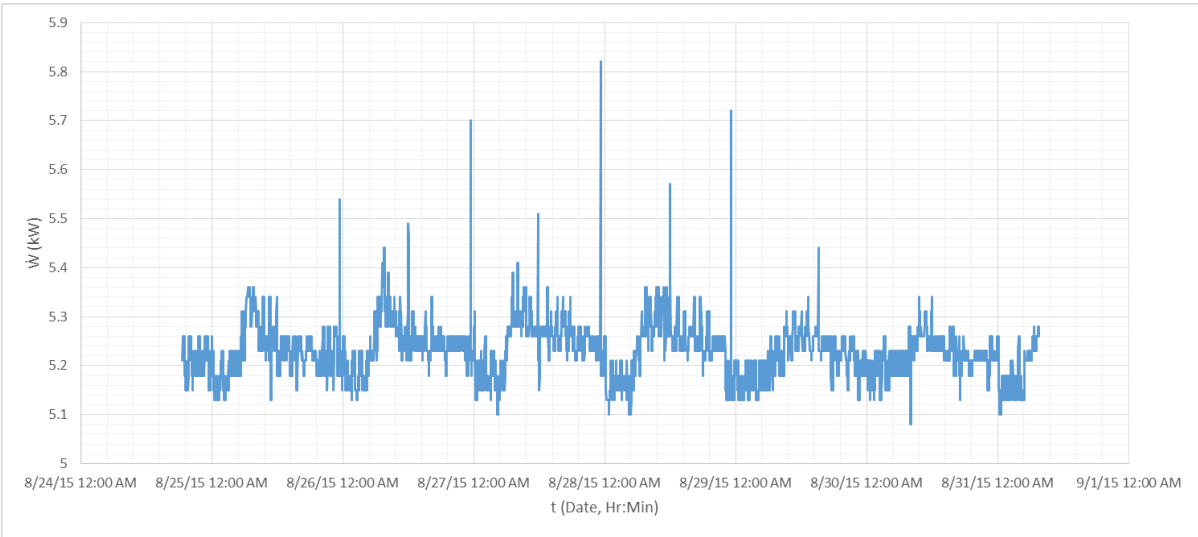


Fig. H20. Power consumption of Worthington pump when it ran from August 24 to August 31 of 2015

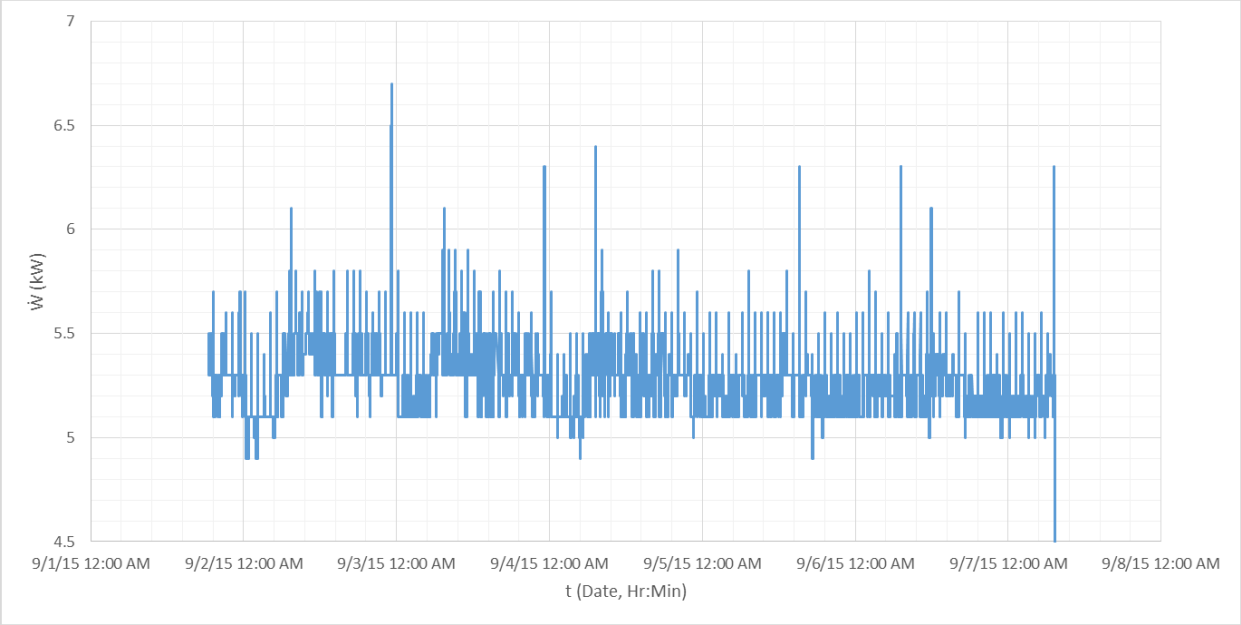


Fig. H21. Power consumption of Grundfos pumps when they ran from September 1 to September 7 of 2015

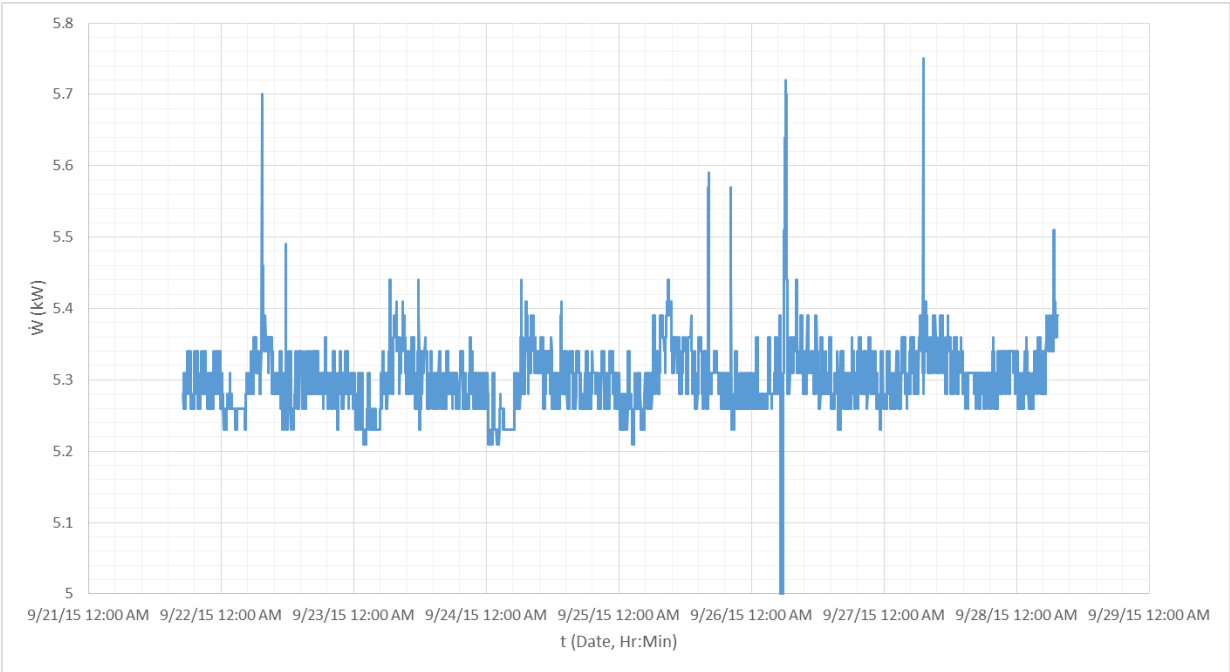


Fig. H22. Power consumption of Worthington pump when it ran from September 21 to September 28 of 2015

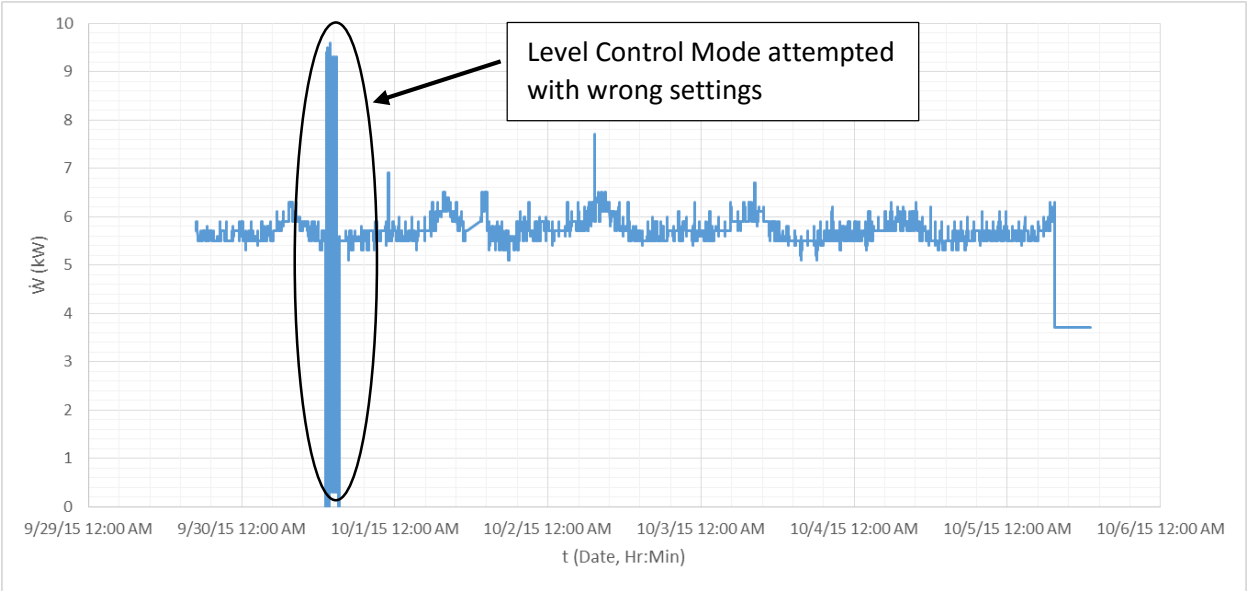


Fig. H23. Power consumption of Grundfos pumps when they ran from September 29 to October 5 of 2015

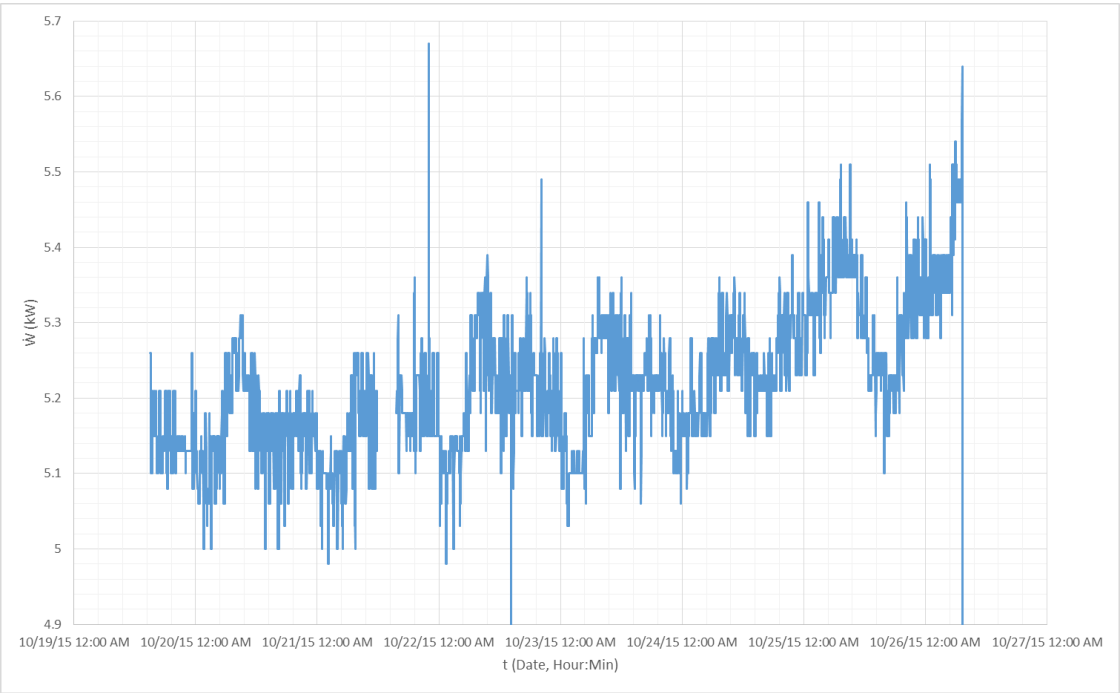


Fig. H24. Power consumption of Worthington pump when it ran from October 19 to October 26 of 2015

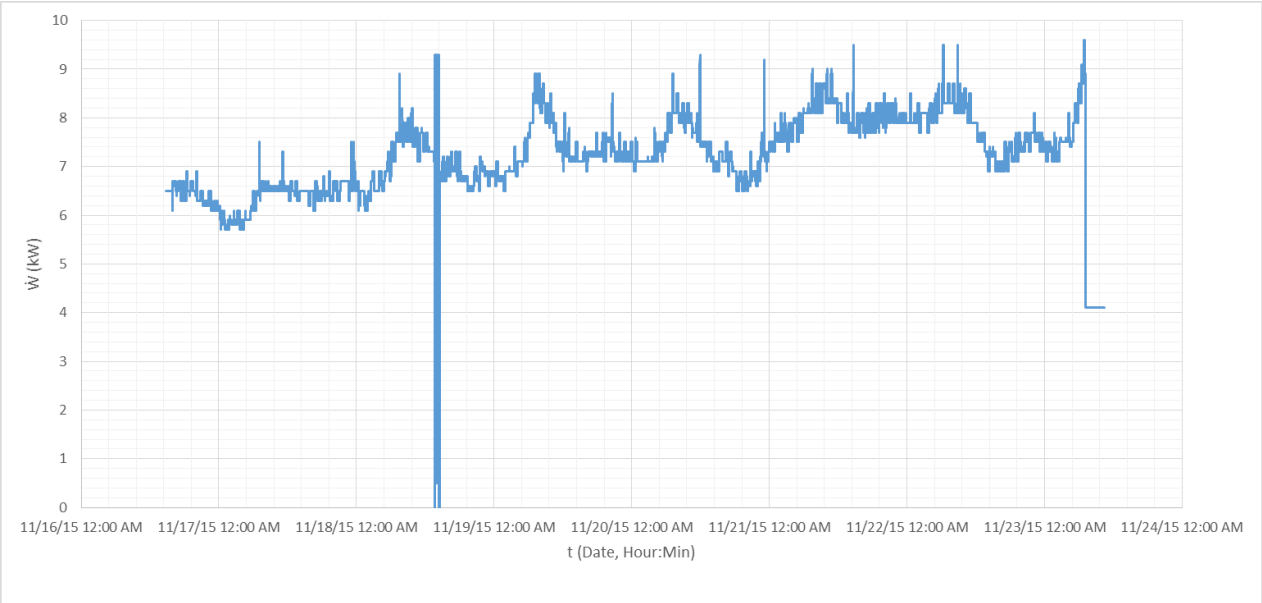


Fig. H25. Power consumption of Grundfos pumps when they ran from November 16 to November 23 of 2015

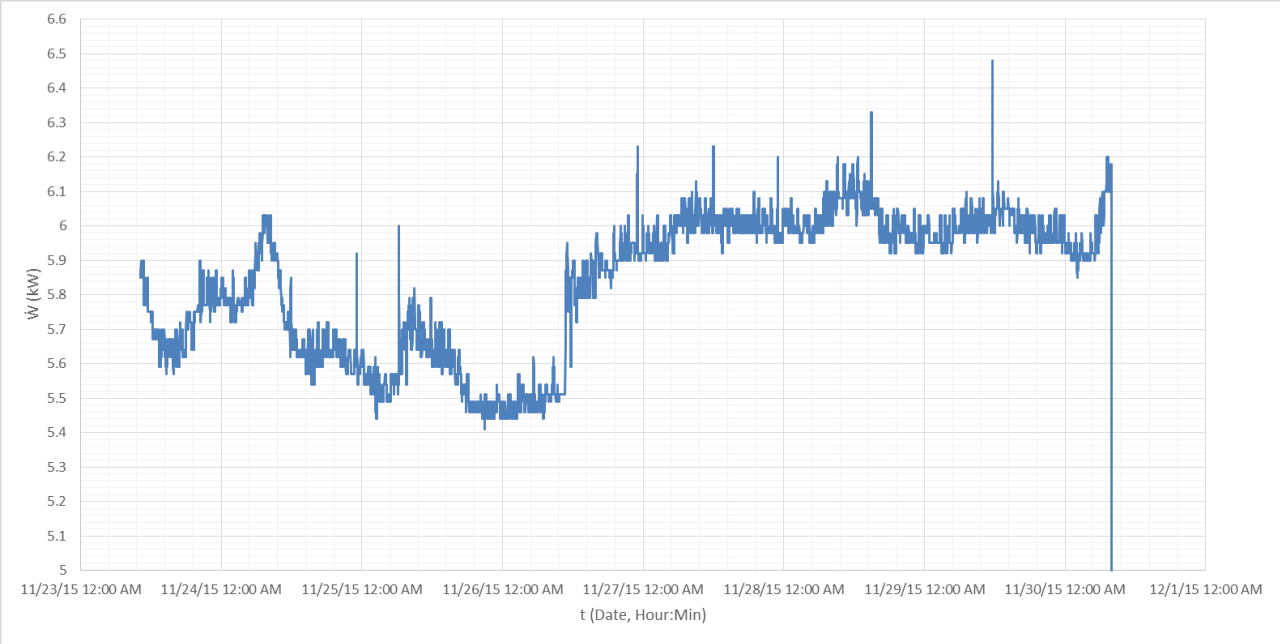


Fig. H26. Power consumption of Worthington pump when it ran from November 23 to November 30 of 2015