Drain Water Heat Recovery Characterization and Modeling

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-Final report-

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Table of Contents

List of Figures

List of Tables

1. Introduction

This study continues the technology assessment of Drain Water Heat Recovery (DWHR) devices initiated in 2005 and presented in *Performance Evaluation of Drain Water Heat Recovery Technology at the Canadian Centre for Housing Technology, C. Zaloum, J. Gusdorf and A. Perekh, 2006*. It was concluded that DWHR devices only recovered energy during simultaneous water draws (showers) and that for modelling purposes all other water draws could be ignored. Based on this conclusion, we sought to characterize the DWHR devices in order to develop an energy savings calculator. The purpose of this experiment is not only to assess the performance of the DWHR units but also to develop a standard tests and modelling methods which would allow the manufacturers, utilities and governments to have reliable data to estimate energy savings and pressure drop.

The DWHR technologies tested are fairly simple in design and can be effective in reducing the amount of energy needed to produce hot wate. The units used in this study have similar designs which consist of 3 inch nominal (76.2 mm) copper drain pipe wrapped with either half-inch nominal (12.7 mm) or 3/8 inch nominal (9.5mm) soft copper tubing, where cold water is circulated recovering heat from the drain. This is shown in figure 1. The DWHR units obtained from the manufacturers are of various lengths and configurations but all have the same basic design.

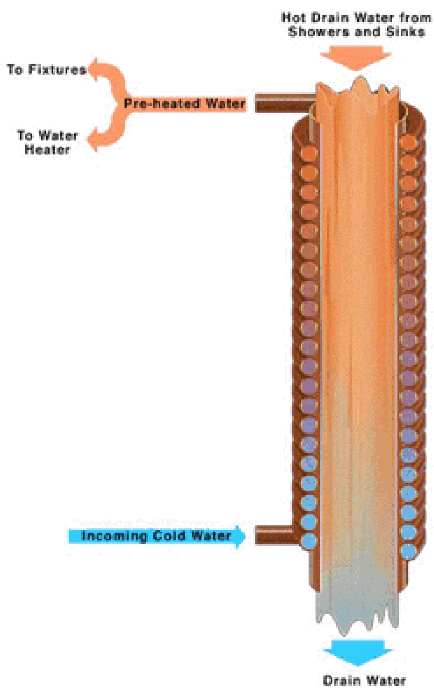


Figure 1: DWHR Principle

Eight different units were tested in this project. The experiments were performed using two different flow configurations, three different flow rates and three different shower temperatures in an effort to assess the performance and heat transfer rate of each unit in comparison with the others. The two different flow configurations change the volumetric flow rate, therefore affecting the heat transfer and performance.

2. Objectives and scope

The main objectives of this study were to:

1- Measure energy recovery and performance with the following configurations
   a. Shower flows of 6.5, 8.5 and 10 Liters per minute
   b. Shower temperatures of 37, 41 and 45 degrees Celsius
   c. Configurations A (pre-heat to hot water tank only) and B (pre-heat to hot water tank and cold water to shower)
   d. Cold water supply fixed at 8 Celsius

2- Establish correlations in order to develop an energy savings calculator

3- Develop the flow vs. pressure loss curves for each unit

4- To develop a test procedure for future performance testing and characterization of DWHR units

The scope of this project was to:

Evaluate and characterize each of the eight DWHR units listed below:

1- GFX G3-60
2- GFX G3-40
3- Power Pipe R3-60
4- Power Pipe R3-36
5- Retherm S3-60
6- Retherm C3-40
7- Watercycles 60
8- Watercycles 36

It is worth noting that, although the units carry the extension "60" or "36" to indicate the unit's length in inches, we found that in most cases it was approximate. Actual pipe lengths and coil lengths are listed in Table 5. We would also like to acknowledge the fact that the Watercycle units submitted for this project were the first three inch diameter units produced by the company. This may be reflected in the results and we would caution against using these results to judge future production units.

3. Setup and Test Procedure

To properly evaluate the various Drain Water Heat Recovery (DWHR) units we needed a controlled environment and standardization of tests. This allows us to confidentially compare performance between units, model performance for a given unit and to assess experimental repeatability.

3.1. DWHR Setup

The DWHR units were installed vertically on the main drain pipe which is a 10 ft. straight run to the shower drain. The units were wrapped with ½ in. closed cell foam insulation to ensure minimal heat loss/gains to the surroundings and reduce the risk of surface condensation. Figure 2 shows the wrapped DWHR unit. Two pressure gages were mounted on the cold water supply, one at the inlet and the other at the outlet, of the outer coil. Pressure transducers were later installed to replace the dial gauges in order to measure the pressure loss in the outer coil more accurately. Thermocouples were installed on the top and bottom ABS pipes and at the inlet and outlet of the outer coil.

Figure 2: Insulated DWHR

3.2. Configurations

The water flow configurations were an important aspect of the experiment as they affect the flow rate going through the heat exchanger.

The experiment was performed using two different water flow configurations as follows:

Configuration A: Route all coldwater going to the hot water tank (HWT) through the Drain Water Heat Recovery (DWHR) Unit only. In other words, the unit preheats the water going to the hot water tank (HWT) only.

Configuration B: Route all cold water flow going to the shower and the hot water tank (HWT) through the Drain Water Heat Recovery (DWHR) Unit. In other words, all the water going to the HWT and shower is preheated.

A third configuration where cold water warmed in the DWHR is routed to the cold water shower tap was not assessed due to resource and time constraints.

In addition to the two flow configuration, shower flow rates were be specified. Tested flow rates were of 6.5 litres/minute, to simulate an ultra-low flow shower head, 8.5 l/m, to simulate low-flow and 10.5 l/m for older style high flow shower head. The tests were also performed with different shower temperatures of 37, 41 and 45 Celsius to simulate cool, warm and hot shower temperatures.

3.3. Chilled Water Setup

Since the city water temperature fluctuates by approximately 13ºC over the course of the year, a method of controlling cold water temperature had to be devised. The national average city water temperature is 8ºC, so the tests needed to be conducted at a constant cold water supply temperature of 8ºC. Given the time of the year and the length of time needed to perform the testing, the water was warmer than the national average.
A system needed to be developed in order to provide chilled water to the house in order to obtain more accurate and repeatable results. The solution to the problem involved using a 2 Kilowatt chiller. Although this chiller was not capable of handling the direct required cooling capacity at the required flow rate, it could cool a large volume of water over a longer period of time. The chiller was used to chill two reservoirs, one of 151L (40 US gallons) and the other of 150 L. Once the two reservoirs were cooled, in a closed loop arrangement powered by a circulating pump, they were connected to the house's water supply. The city water would enter the top of one tank and push the cold water out the bottom then to the top of the second tank (See figure 3). A cold water mixing valve was installed between the cold and city flow, which allowed the tanks to be cooled to a lower temperature and also have better temperature control.

Figure 3: DWHR test setup

3.4. Pressure testing

The static pressure loss across the DWHR units was measured using pressure transducers which were connected to the data logger. The pressure losses were measured at 6 L/min, 8L/min, 10 L/min, 12L/min and at maximum flow.

3.5. Data Acquisition and Measurements

To measure several data points in the experiment a Campbell CR10x system was used, and this was connected to 8 type K thermocouples, 2 type T thermocouples, 3 water meters, and a relay to control the shower. The data was then downloaded to a computer and analyzed in a spreadsheet. Pressure gauges were also installed at the top and bottom of the DWHR cold flow pipe, and results were verified using a Fluke 789 process meter equipped with a pressure transducer.

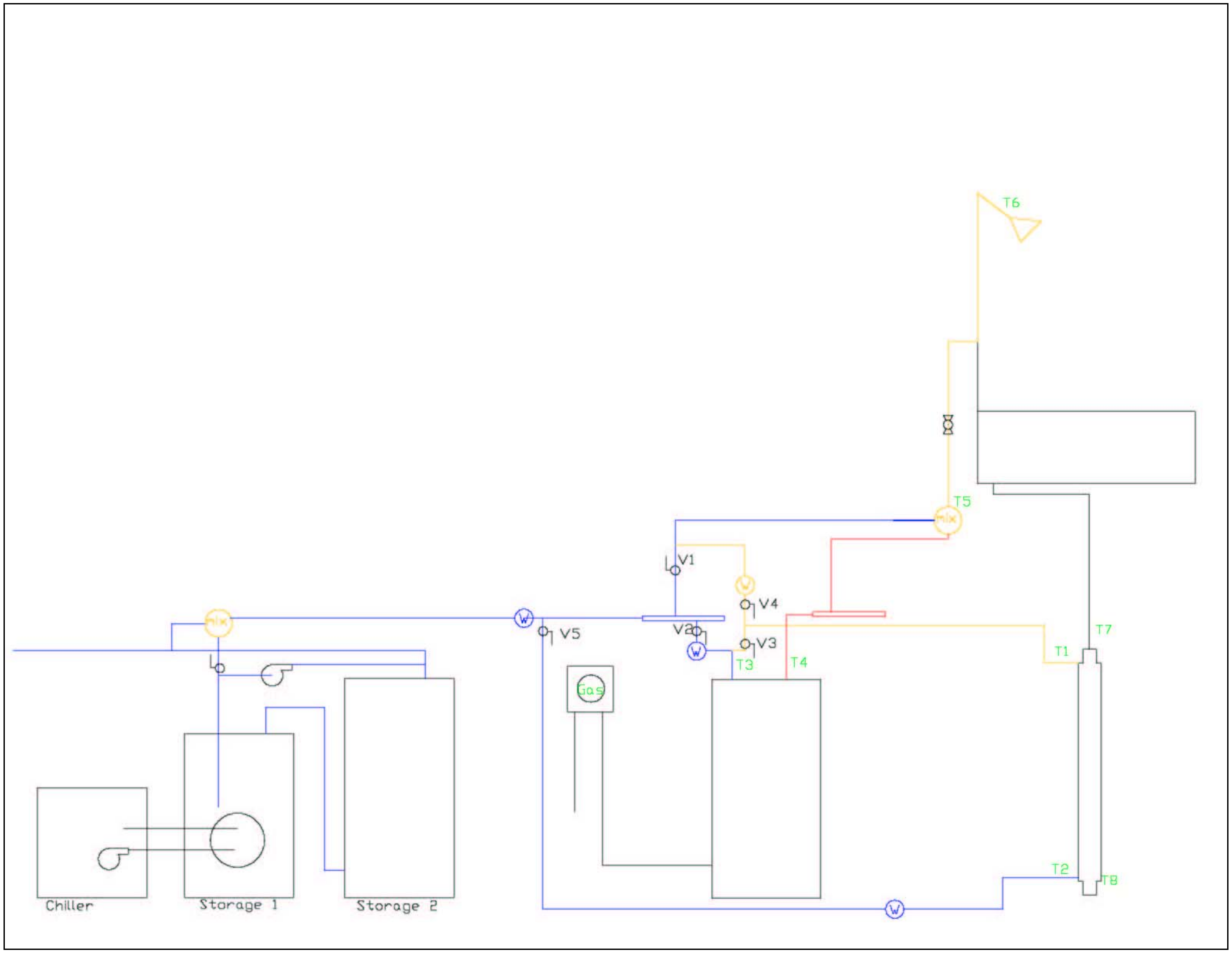


Figure 4: Plumbing and instrumentation layout

3.6. Test Procedure

Initially, the impact of all parameters (flow, temperature, configuration A & B) needed to be evaluated with 18 tests per DWHR unit. After the tests were performed on the GFX series it was found that shower temperature had no observable impact on unit performance. As a result, the number of tests could be reduced by half (8 to 10 tests); clear trends were evident in the calculated NTU, which is partially based on flowrates. It was observed that for a given DWHR unit the only relevant parameter to determine the NTU for the heat exchanger was the flow rates. This could be done as long as a sufficient flow range from roughly 4L/min to 10.5L/min could be obtained. Table 1 shows which tests were performed on each unit.

Configuration A and B were used to maintain consistency with the experimental setup once the NTU correlation was determined.

A simplified test procedure can be developed in the future by keeping the configuration and shower temperature constant while changing flow rates. Initially the test showers were run for 30 minutes, it was then found that this could be reduced to 15 minutes because steady state was always reached well within 15 minutes.

Table 1: Flow Configurations

	GFX 60	GFX 40	Power Pipe 60	Power Pipe 36	Retherm 60	Retherm 40	Watercycles 60	Watercycles 36
Config A								
6.5 L/min								
37C								
41C								
45C								
8.5 L/min								
37C								
41C								
45C								
10.5 L/min								
37C								
41C								
45C								

	GFX 60	GFX 40	Power Pipe 60	Power Pipe 36	Retherm 60	Retherm 40	Watercycles 60	Watercycles 36
Config B								
6.5 L/min								
37C								
41C								
45C								
8.5 L/min								
37C								
41C								
45C								
10.5 L/min								
37C								
41C								
45C								

	Tested Twice

Calculations

In order to calculate the performance and the amount of heat transfer for a particular heat exchanger there are two methods, the logarithmic mean temperature difference (LMTD) and Number of heat transfer units (NTU)-effectiveness. Both were used during the calculations and compared, although it was concluded that the NTU was more appropriate since it does not require exit temperatures. Both methods are shown below.

For more details refer to the *2005 ASHRAE Handbook-Fundamentals, page 3-28*. The empirical heat transfer rate coefficient was solved using the actual effectiveness derived from the NTU method as shown below. A fully detailed calculation is included in Appendix 1.

3.7. Logarithmic Mean Temperature Difference

The logarithmic mean temperature difference is a measure of the temperature difference along the heat exchanger. The temperature distribution of a counter flow heat exchanger is shown in fig 4.

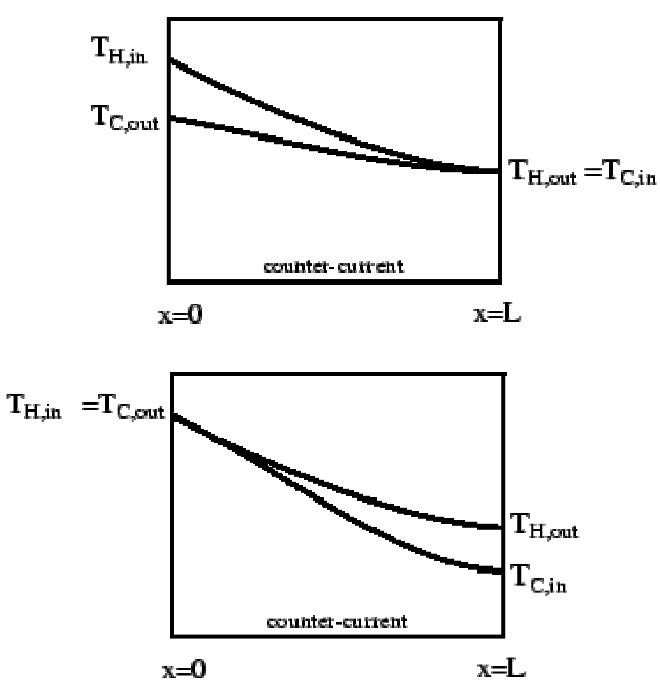


Figure 5: Counterflow HE Temperature Distribution

LMTD counterflow heat exchanger

$$\Delta T_{lm} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln\left[\frac{(T_{hi} - T_{co})}{(T_{ho} - T_{ci})}\right]} \quad ^{o}C$$

$$q = UAF\Delta T_{lm} \quad \text{kW}$$

UA = Overall heat transfer coefficient kW/ºC

F = Correction factor

It is to be assumed that the correction factor F is 1.0 or near 1.0 due to the fact that it is technically a counter flow heat exchanger, with one shell and one tube pass, although it is configured differently than a conventional unit. The exception is the split DWHR unit, like the Retherm 60, which should technically be modeled as two counter flow heat exchanger as there are two separate cold water coils connected in series.

The only downside to this method is that it requires the knowledge of the outlet temperatures which, in most cases, are unknown when trying to predict the performance of a heat exchanger with minimal input.

3.8. NTU-Effectiveness

The second method of calculating the overall heat transfer coefficient and the rate of heat transfer is by the Number of heat transfer units (NTU)-effectiveness method. The NTU is a measure of the heat transfer size of the heat exchanger; the larger the NTU the closer the heat exchange approaches its thermodynamic limit. The effectiveness is the ratio of the actual rate of heat transfer to the maximum possible rate of heat transfer in a heat exchanger. This method is particularly useful when outlet temperatures are not given.

$$NTU = \frac{UA}{C_{\min}}$$

$C_{\min}$ The lesser of $(\dot{m} c_p)_{cold}$ or $(\dot{m} c_p)_{hot}$.

In evaluating configurations A and B, C_{min} is always on the cold side.

$$= (\dot{m} c_p)_{cold} \quad \text{KJ/s ºC}$$

$C_{\max}$	The greater of $(\dot{m}c_p)_{cold}$ or $(\dot{m}c_p)_{hot}$. In evaluating configurations A and B, C_{max} is always on the hot side. $= (\dot{m}c_p)hot$ kJ/s ºC
C_r	$= \dfrac{C_{\min}}{C_{\max}}$
$\dot{m}$	= Cold water mass flow rate kg/sec
c_p	= Fluid heat capacity of cold water kJ/kg ºC

The empirical heat transfer rate is calculated using the actual effectiveness which takes into account the outlet temperature of the cold fluid.

$$\varepsilon = \frac{\left(T_{c,o} - T_{c,i}\right)}{\left(T_{h,i} - T_{c,i}\right)}$$

$$q = \varepsilon C_{\min}\left(T_{hi} - T_{ci}\right) \qquad \text{kW}$$

The NTU can then be determined using the actual effectiveness and the specific heat ratio.

For a counterflow heat exchanger

Configuration A

$$NTU = \frac{1}{\left(C_r - 1\right)}\ln\left(\frac{\varepsilon - 1}{\varepsilon C_r - 1}\right) \qquad \left(C_r \neq 1\right)$$

Configuration B

$$NTU = \left(\frac{\varepsilon}{1 - \varepsilon}\right) \qquad \left(C_r \neq 1\right)$$

Once the NTU is obtained the overall heat transfer coefficient UA can be calculated

$$UA = NTU \cdot C_{\min} \qquad \text{kW/ºC}$$

Using spreadsheet software that permits curve fitting or a numerical method, a standard exponential NTU vs. Flow rate curve can be correlated for each unit due to the consistent relationship between the NTU and the flow rate. These equations can be used to predict the amount of heat transfer between the fluids, without having to test the unit. All that is required to work the problem backwards to solve the heat transfer rate are the inlet temperatures, the NTU curves, flow rate and the type of unit. The NTU equations for each unit are tabulated in section 4.2.1.

3.9. Theoretical Calculation

Once the curve for the NTU is obtained the problem can be solved backward to determine the theoretical NTU, effectiveness and heat flow (example calculations in Appendix 3).

NTU = Specific equation for each unit

Working out the theoretical effectiveness with the calculated NTU

$$\varepsilon = \frac{1-\exp\left(-NTU(1-C_r)\right)}{1-C_r\exp\left(-NTU(1-C_r)\right)} \qquad (C_r \neq 1)$$

$$C_r = \frac{C_{\min}}{C_{\max}}$$

$$\varepsilon = \frac{NTU}{1+NTU} \qquad (C_r = 1)$$

By solving for effectiveness we can find the heat flux with the inlet temperatures.

$$q = \varepsilon C_{\min}\left(T_{hi} - T_{ci}\right) \qquad \text{kW}$$

$$C_{\min} = (\dot{m}\, c_p)_{cold} \qquad \text{kJ/s ºC}$$

$\dot{m}$ = Mass flow rate (use hot water flow for config A, total flow for config B) kg/sec

c_p = Fluid heat capacity kJ/kg ºC

3.10. Pressure Drop

The Bernoulli equation is used to analyze fluid flow from one location to another. In order to use this equation, four criteria must be met; the fluid must be incompressible, the fluid must maintain a consistent density, the flow must be steady, the flow must be along a streamline. These conditions are met in this case; therefore, the following equation will be used to determine the flow coefficient to be used in future analysis and/or modelling.

Pressure drop (psi)

$\Delta P \quad = A\,Q^2$

Where, A is the flow coefficient
Q is the water flow (L/min)

4. Results and Data Analysis

The experimental results and data analysis are presented below.

4.1. Controlled and Uncontrolled Variables

Although the tests were structured in such a way as to ensure repeatability and standardization, some uncontrollable variables could have affected the results. These include variables such as the ambient temperature, water flow rate, shower temperature, etc.

The shower flow rate is subject to fluctuations due to city water pressure and meter accuracy, an error of plus or minus 0.5 L/min is expected. The flow rate was controlled using a ball valve, as controlling the flow using different shower heads not sufficiently precise. Water pressure is also subject to variation as other building on the NRC complex draw large amounts of water, thus reducing overall pressure and affecting the flow. Since the available water meters only generate one pulse per liter, the flow rate was measured by counting the pulses within an interval of one minute. The flow was then averaged over the length of the test.

The shower temperature was controlled using a thermostatic mixing valve, which was adjusted manually, and this may also have contributed to some slight variations. The shower water mixing temperature was observed and adjusted as needed, though it was known to not be an important factor. A cold water mixing valve was also installed between the chilled water and city water lines in an effort to dampen any variation in the cold water temperature.

Since it was determined that the each DWHR unit's heat transfer performance could be characterized by a NTU vs. Flow rate curve, we knew the uncontrolled variables did not have a significant effect. The key variable: flow rates, inlet temperatures and outlet temperatures were the important variables. These were easily measured and were not affected by external influences. NTU could still be calculated even if there was slight inconsistency in the variables. For more details refer to the calculations section.

4.2. Unit Performance

The performance of each unit was measured in terms of the Number of Transfer Units (NTU) and effectiveness. The NTU seemed to show a better correlation factor than that of the effectiveness. The NTU is a measure of the heat transfer size of the heat exchanger. The larger the NTU the more the heat exchanger approaches its thermodynamic limit. On the other hand the effectiveness is the ratio of the actual rate of heat transfer to the maximum possible rate of heat transfer. The actual effectiveness curves for the heat exchangers didn't seem to correlate as well as for the NTU method, R^2 being the correlation coefficient. Analysis was done for both minute 1 through 11 and 5 through 11. The idea was that minutes 1 through 11 would be a more realistic simulation as the heat exchanger goes through the transient (warm up) stage, whereas the 5-11 minutes was to simulate a steady state operation. The data is also presented on a per foot basis using the correlated equations.

4.2.1. NTU

Figure 5 from minute 1 to 11 shows the NTU curves fitted to the empirical data. It can be seen that the best performing unit is the Power Pipe 60 followed by the GFX 60 and so on. The 5 to 11 minute curves are included in appendix 3. The equations are summarized in table 2 for both minutes 1 to 11 and 5 to 11.
It can be observed that there is a lot of scatter for the Watercycles 36; this may be caused by the shortness or design of the pipe.

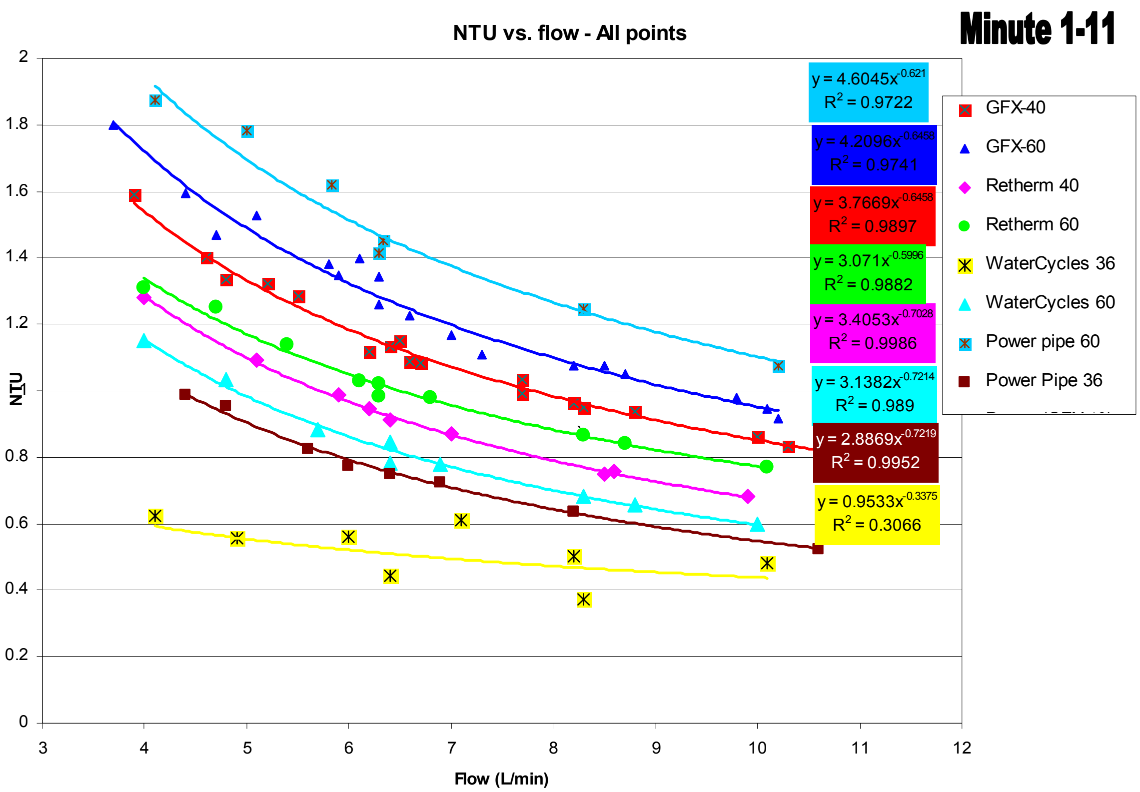


Figure 6: NTU Curves (Minutes 1-11)

Table 2: NTU equations

Manufacturer	Model	Minutes 1-11		Minutes 5-11	
		NTU-Equation	R^2	NTU-Equation	R^2
GFX	G3-40	$y=3.7669x^{-0.6452}$	0.9897	$y=4.0168x^{-0.6678}$	0.9946
GFX	S3-60	$y=4.2096x^{-0.6458}$	0.9751	$y=4.4495x^{-0.6711}$	0.9831
Retherm	C3-40	$y=3.4053x^{-0.7028}$	0.9986	$y=3.3371x^{-0.6817}$	0.9969
Retherm	S3-60	$y=3.071x^{-0.5996}$	0.9882	$y=3.1314x^{-0.713}$	0.9899
Power Pipe	C3-36	$y=2.8869x^{-0.7219}$	0.9952	$y=3.0514x^{-0.7401}$	0.9981
Power Pipe	R3-60	$y=4.7622x^{-0.6355}$	0.9666	$y=5.0866x^{-0.6601}$	0.9727
Watercycles	36	$y=0.9533x^{-0.3375}$	0.3066	$y=0.98x^{-0.3465}$	0.3141
Watercycles	60	$y=3.1382x^{-0.7214}$	0.989	$y=3.1314x^{-0.713}$	0.9962

The NTU can also be observed on a per foot basis using an arbitrary flow rate value of 8.5 L/min. It can be seen that the GFX 40 unit performs best on a per foot basis followed by the Power Pipe 60. The per foot basis may be important if the units alone are looked at from an economical stand point as copper prices have risen significantly in the last decade. Figure 6 shows the comparison between the units. It can also be observed that there is a tendency for the longer pipes to have a lower NTU/foot, this is because as the two fluid temperatures approach each other the amount of heat transfer diminishes, which also affects the NTU on a per foot basis.

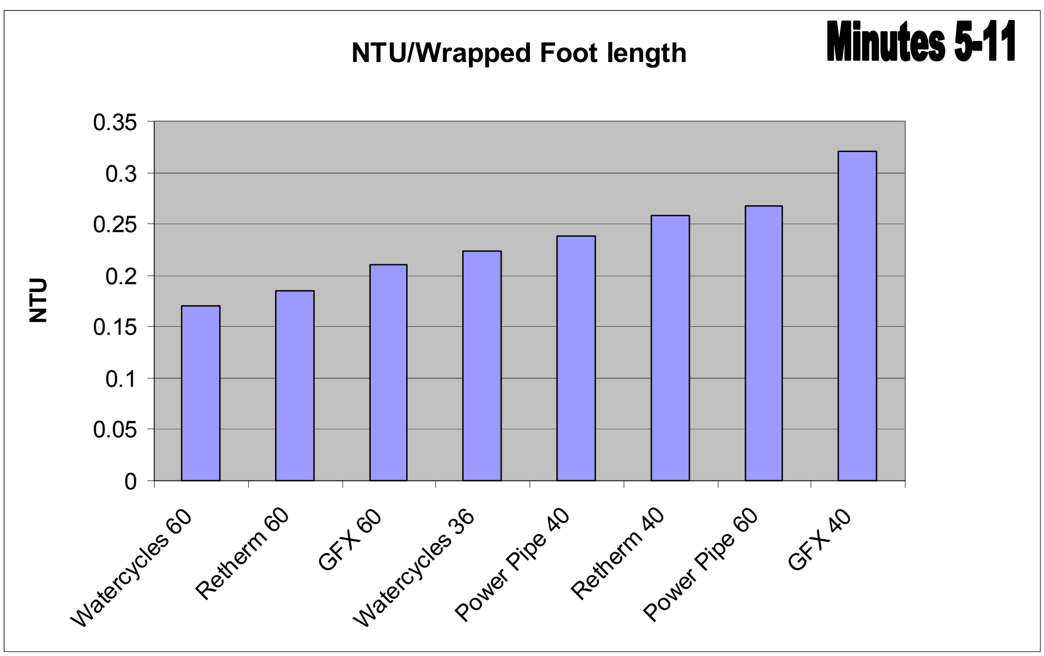


Figure 7: NTU/ Wrapped Foot Length

4.2.2. Effectiveness

The effectiveness of the different units from minute 1 to 11 can be observed in figure 7. It can be seen that the best performing unit is again the Power Pipe 60 followed by the GFX 60. The effectiveness curves follow the same order as the NTU, as they are related to each other. The equations are tabulated in Table 3.

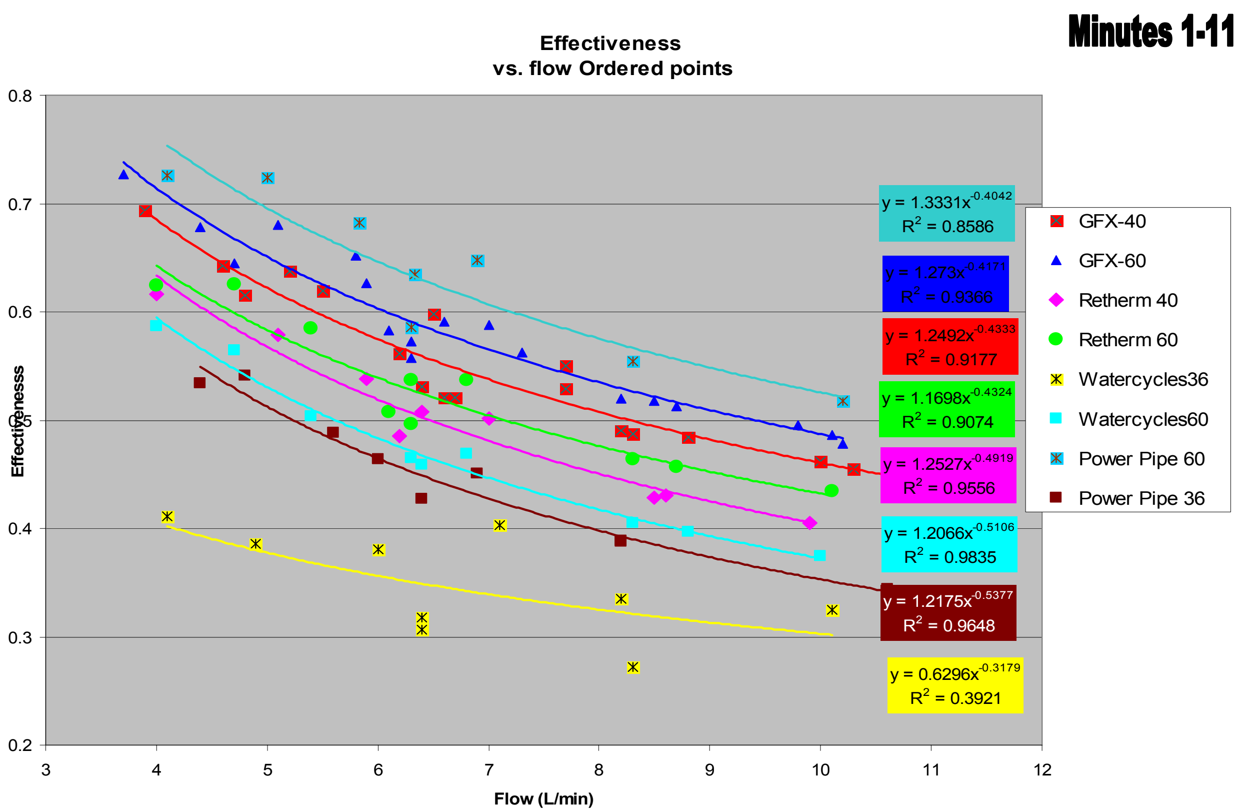


Figure 8: Effectiveness Curves (Minutes 1-11)

Table 3: Effectiveness Equations

Manufacturer	Model	Minutes 1-11		Minutes 5-11	
		Eff-Equation	R^2	Eff-Equation	R^2
GFX	G3-40	$y=1.245x^{-0.4328}$	0.8899	$y = 1.245x^{-0.4328}$	0.8899
GFX	S3-60	$y=1.2976x^{-0.4278}$	0.9185	$y = 1.2976x^{-0.4278}$	0.9185
Retherm	C3-40	$y=1.2527x^{-0.4919}$	0.9556	$y = 1.2204x^{-0.4714}$	0.959
Retherm	S3-60	$y=1.1698x^{-0.4324}$	0.9074	$y = 1.1872x^{-0.4338}$	0.908
Power Pipe	C3-36	$y=1.2175x^{-05377}$	0.9648	$y = 1.2593x^{-0.5484}$	0.9718
Power Pipe	R3-60	$y=1.3331x^{-0.4042}$	0.8586	$y = 1.3575x^{-0.4091}$	0.8672
Watercycles	36	$y=.6296x^{-0.3179}$	0.3921	$y = 0.6488x^{-0.3284}$	0.4001
Watercycles	60	$y=1.2066x^{-0.5106}$	0.9835	$y = 1.2313x^{-0.514}$	0.9728

The effectiveness can also be looked at on a per foot basis like the NTU which gives a better understanding of the marginal gain a larger pipe offers in comparison to a small pipe. Figure 8 shows the comparison between the different pipes. It can be seen that in this case the GFX 40 also dominates followed by the Watercycles 36, but the order differs slightly from that of the NTU/foot. They do seem to follow the same kind of pattern where the 60 inch pipes have lower values of effectiveness. The reason being is that as the two fluid temperatures approach each other the less there is heat transfer, therefore it only becomes marginally more effective to have a longer pipe, thus reducing the effectiveness on a per foot basis.

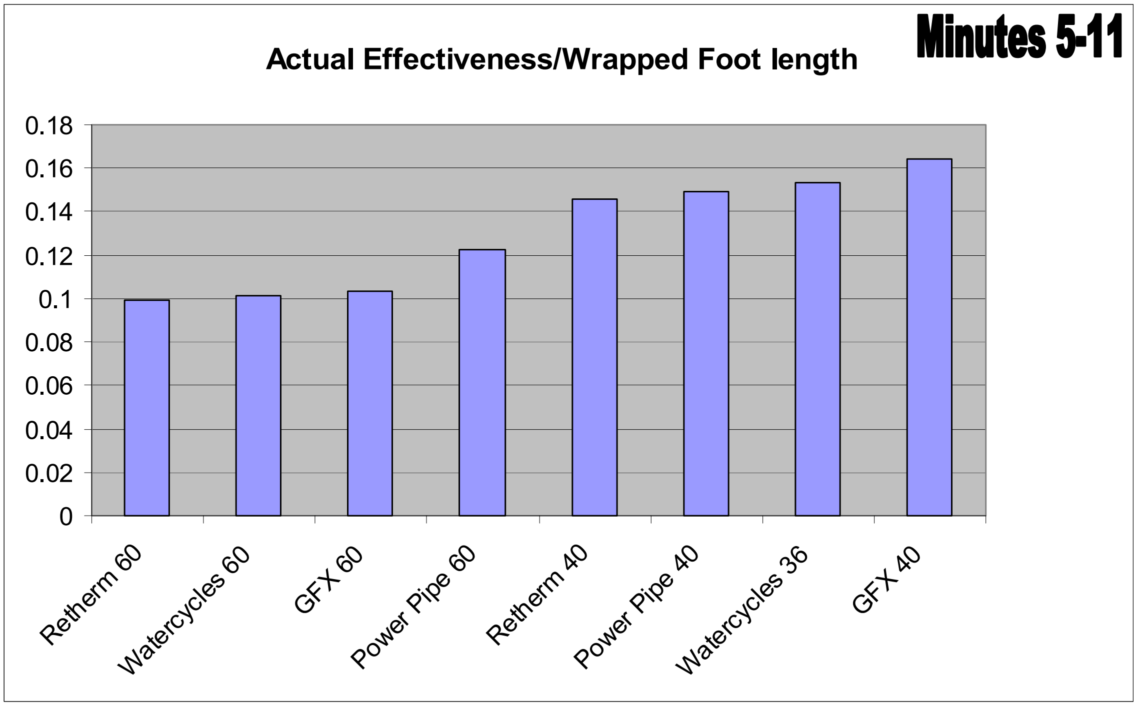


Figure 9: Actual Effectiveness/Wrapped Foot Length (Minutes 5-11)

4.3. Pressure Drop

Static pressure loss across each DWHR unit was measured for each pipe at different flow rates. Excessive pressure loss in the pipes can affect water flow rate and lead to consumer complaints. The pressure loss was measured at 6 L/min, 8L/min, 10 L/min, 12L/min and at maximum flow. Figure 9 shows the pressure difference vs. flow rate. It can be observed that the GFX G3-60 unit creates the greatest pressure loss, which is also the longest unit. The Power Pipe units on the other hand have a smaller pressure loss due to their design where water simultaneously travels through the four 3/8" tubes, instead of one ½" tube. The Retherm S3-60 also has low pressure loss due to its design that splits the flow into two sections. Table 4 shows the correlated equations for the pressure loss.

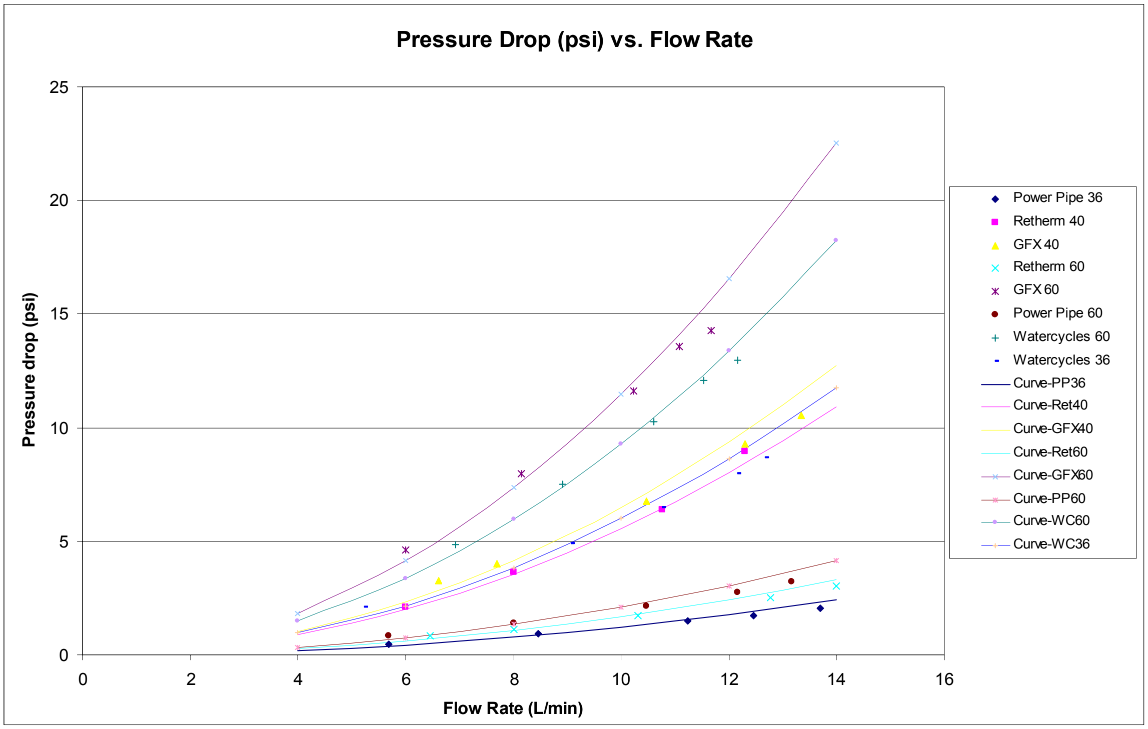


Figure 10: Pressure Drop vs. Flow Rate

Table 4: Pressure Drop Equations

Manufacturer	Model	Pressure Drop Equation
Power Pipe	R3-60	$y = 0.021116\ x^2$
Power Pipe	R3-36	$y = 0.012256\ x^2$
GFX	G3-60	$y = 0.114931\ x^2$
GFX	G3-40	$y = 0.064957\ x^2$
Retherm	C3-40	$y = 0.055603\ x^2$
Retherm	S3-60	$y = 0.016998\ x^2$
Watercycles	60	$y = 0.09297\ x^2$
Watercycles	36	$y = 0.059983\ x^2$

Figure 10 shows the Pressure Difference vs. Flow Rate on a per foot of coiling basis, it can be observed that most units have a similar per foot pressure loss except for the Power Pipe which has low flow resistance.

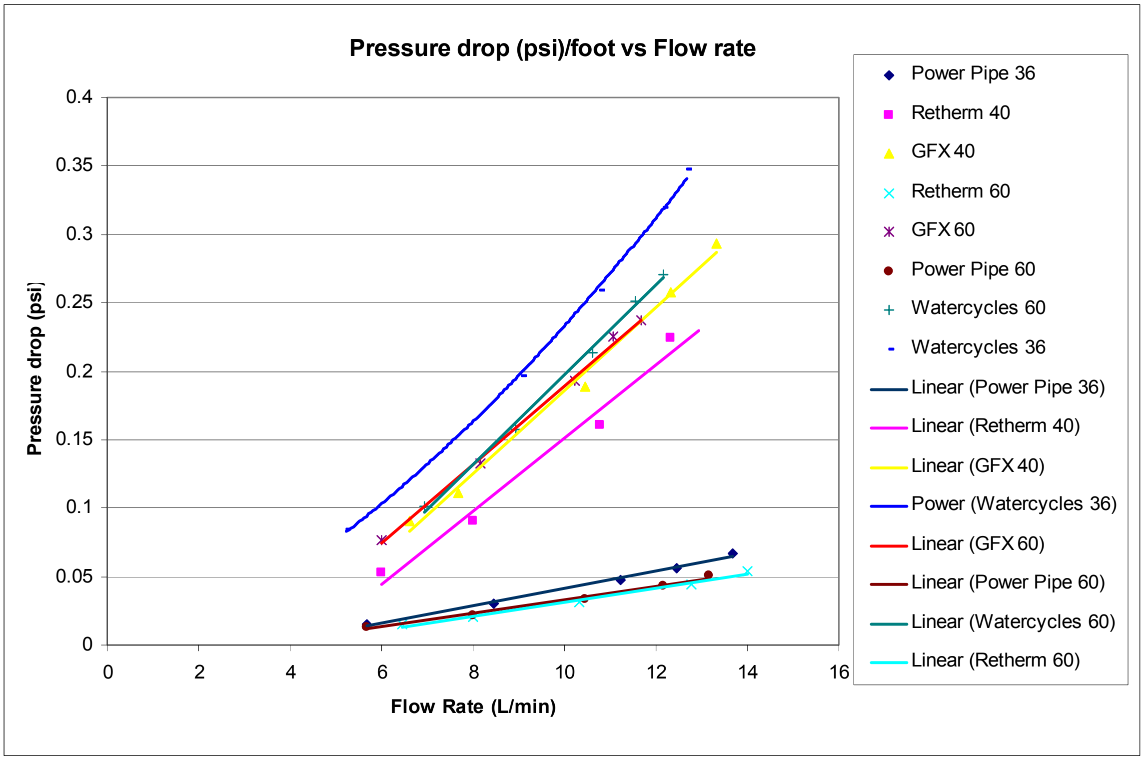


Figure 11: Pressure Drop(psi)/Foot vs. Flow Rate

4.4. Predicted NTU and Heat Flow

Once the correlations were determined from the empirical data, the problem had to be solved backward in order to validate the equations. The calculation involved solving for the NTU from the equations using the flow rates. The effectiveness was then solved from the NTU and using the inlet temperatures and the specific heat the amount of heat transfer was determined. Figure 11 shows the actual and theoretical heat transfer vs. heat flow curve for the GFX 40 pipe. Theoretical results were calculated using the NTU curve-fitted equations.

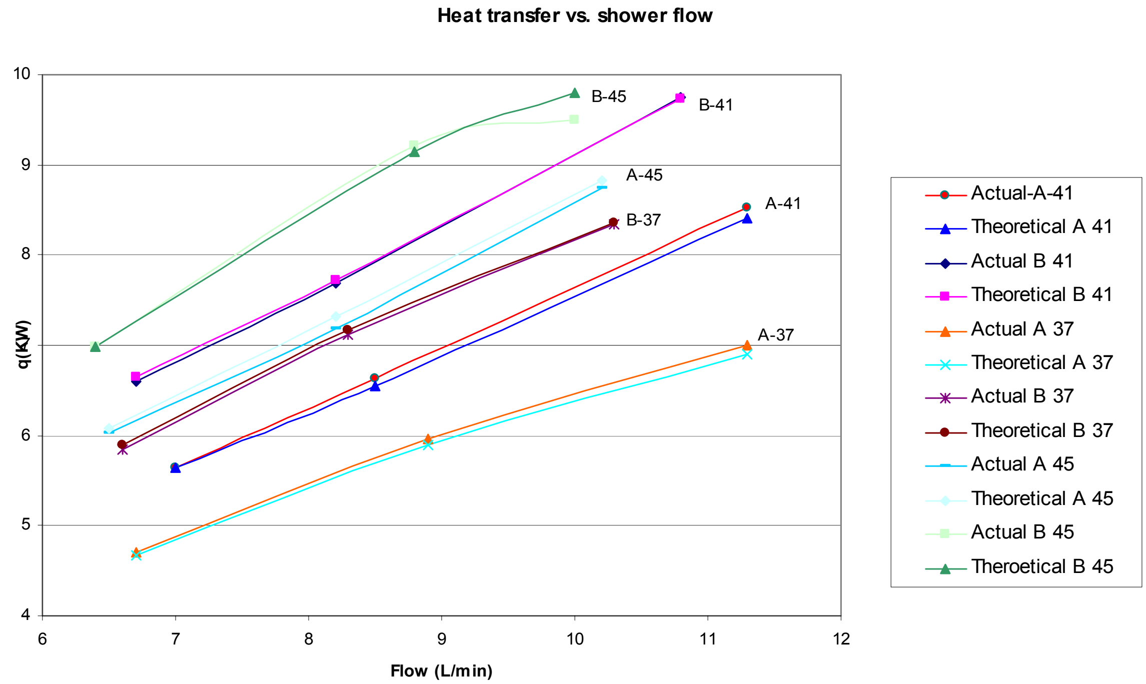


Figure 12: Actual and Theoretical Heat Transfer for configurations A and B at 37, 41 and 45 C shower temperature (B-45 means configuration B with a 45C shower temperature)

5. Discussion

5.1. DWHR Unit Specifications and Differences

Numerous geometrical and design aspects were observed in order to determine if they could be related to performance. It seemed the smaller the gap between the tube coiling the higher the performance, and also the lower the vertical flow rate the higher the performance. What is meant by vertical flow rate is that for every rotation around the pipe the fluid moves vertically by the center to center distance between the tubing (see Figure 12).

The Power Pipe 60 was among the highest in performance due to its design which has four smaller 3/8”tubes simultaneously wrapped around the 3” pipe rather than a single ½” tube wrapping. On the other hand the Power Pipe 36 is among the least performing; therefore it may only be a suitable design for longer pipes only. Table 5 shows the geometric differences between the DWHR units.

Figure 12: From left to right; Retherm, Watercycles, GFX, PowerPipe

Table 5: Geometric details

Manufacturer	Model	Actual Length(in)	Coiling Length(in)	Inner Pipe Dia. (in)	Tubing Dia(in)
Power Pipe	R3-60	64	55.5	3	0.375
Power Pipe	R3-36	36	31		0.375
GFX	G3-60	65	60.25	3	0.5
GFX	G3-40	40	36	3	0.5
Retherm	C3-40	40	36	3	0.5
Retherm	S3-60	60	28 x 2	3	0.5
Watercycles	60	56	48	3	0.5
Watercycles	36	29	25	3	0.5
Manufacturer	**Model**	**Tube Passes**	**Winding**	**Squareness**	
Power Pipe	R3-60	Quadruple Pass	Single Sections	Squarest	
Power Pipe	R3-36	Quadruple Pass	Single Sections	Squarest	
GFX	G3-60	Single Pass	Single Sections	2nd Squarest-Nearly Square	
GFX	G3-40	Single Pass	Single Sections	2nd Squarest-Nearly Square	
Retherm	C3-40	Single Pass	Single Sections	3rd Squarest-Nearly Square	
Retherm	S3-60	Single Pass	Two Equal Sections	3rd Squarest-Nearly Square	
Watercycles	60	Single Pass	Single Sections	Rectangular-roundish	
Watercycles	36	Single Pass	Single Sections	Rectangular-roundish	

Looking at the GFX 40 and the Retherm 40, both designs are virtually identical but the GFX 40 outperforms the Retherm 40. It is speculated that the difference may be due to the "squareness" of the tubes; the squarer the tube the more surface contact area it would have with the pipe, thus increasing the heat transfer area. As seen in table 6 the GFX 40 has a smaller outer circumference suggesting a possible tighter coil wrapping than that of the Retherm 40. If a unit has a larger air gap between the tubing and the pipe it would affect the heat transfer rate due to the fact that air is not a good conductor, this effect can be reduced by ensuring a tighter wrap and a squarer tubing, which was most likely the case when comparing the GFX 40 to the Retherm 40.

Table 6: DWHR outer circumference

Model	outer circumference(in)
GFX 60	13.5
GFX 40	13.125
Retherm 40	13.25
Retherm 60	13.25
Power Pipe 60	11.625
Power Pipe 36	11.625
Watercycles 60	12
Watercycles 36	12

The Watercycles pipes, which did not perform very well, have a high vertical flow rate and their tubing wasn't very square and more of a roundish rectangular shape with the width (longer side of the rectangle) of the tube against the pipe. The way the Watercycles pipes were manufactured, most likely by rolling the tube through a single set of rollers, caused a cavity in the centre of the tubing, creating an air pocket which impedes heat transfer flow rate, as air is a poor conductor.

The Retherm 60 differed in its design as it had the cold flow rate split into two different sections, like two separate units connected in parallel on the same pipe. It didn't appear to have a significant gain over its counter part the Retherm 40 which performed almost as well with just a single wrap. With the cold flow splitting into two, one flow going to the upper part of the pipe and one to the lower part, would not have great benefits as the hot water would have been cooled down by the first section of the heat exchanger before passing to the second therefore reducing the performance of the lower part in terms of effectiveness and NTU. This design gained an advantage in the pressure loss test where there was minimal pressure loss over the length of the pipe.

In the GFX pipes the marginal benefit of having a longer pipe could in fact be observed. The overall performance difference between the two was fairly close with the GFX 60 performing slightly better that the GFX 40 as shown in figure 5. On the other hand looking at the two on a per foot basis in terms of NTU and effectiveness, the GFX 40 is superior to the GFX 60. Now comparing the two on a geometric standpoint the GFX 60 has 60.25 inches of tube wrapping almost double the amount of the GFX 40 which has 36 inches of wrapping, this indicates that there is marginal benefits in having additional wrapping length. It can also be observed that the GFX 60 has a larger outer diameter again suggesting the GFX 40 could have a tighter wrap. It has come to our attention that the GFX-60 and GFX-40 did not come from the same manufacturer. The GFX-40 was produced by the current licensee.

5.2. Pressure loss

The pressure loss data presented in the previous section shows that the different pipe configurations can affect the pressure. Pressure loss in the DWHR can become important for low pressure systems, for example a DWHR unit that is connected to a system with a deep well pump which may not be capable of high pressure flows might significantly affect shower pressure. This is where design considerations such as with the Power Pipe and Retherm 60 become useful in reducing pressure loss. This could be advantageous for those in rural areas that don't have access to the city water supply. During this experiment where city water was fed to the system, no significant reduction in water flow was observed at the shower.

5.3. Transient and Steady State

In the previous section the data was analyzed with the initial transient (minutes 1-11) and fully steady state (minutes 5-11). The data was studied in order to observe if there was a significant difference between the two sets of results. The concern was that a longer pipe would take a longer time to reach steady state than a smaller pipe. It was determined that there was only a slight variation between the transient and steady state data.

5.4. Standard Testing

NTU-curves for each pipe must be developed to do energy saving calculations. A simple test to measure the input-output temperatures at representative flow rates is all that is needed. It was found that configurations A and B did not have any effect on developing NTU-curves and that only the flow rate going through the heat exchanger did. Therefore a standard test can be developed even if the water temperatures are not well controlled; as long as the inlet and outlet temperatures and flow rates are known, the NTU vs. flow rate curve can be developed.

In this study, a set of nine tests was used to develop an excellent NTU-curve. With curve-fit NTU-equations, predicting the behavior of the DWHR units, including the effectiveness and amount of heat transfer, is quite simple.

A simpler set-up and data-acquisition system than the one used in this study could be developed for standard testing.

5.5. Development of a DWHR energy savings calculator

Using the simple NTU-curves, developed as part of this project, and applying other theoretical calculations presented previously in this report, a simple energy saving calculator can be developed so that consumers and energy utilities can evaluate the benefits of this technology.

5.5.1. User input requirement

The user would have to input the following information or select from a list.

1. Shower temperature (Ts); select from list;
   Cool (37C)
   Warm (41C)
   Hot (45C)

2. Length of showers (L); input in minutes.

3. Number of showers per day (N); input number

4. Type of shower head (F),

   Low flow (6.5 L/min),
   Standard (9.5L/min)
   Older (15 L/min)
   High flow (18 L/min)

5. Type of hot water tank (DHWt, type) (DHWe, efficiency);

   This is required in order to adjust the savings in light of the systems efficiency. Recovery efficiency is used.

DHW system type	Recovery efficiency (%)
Standard Natural Gas tank	78
High efficiency Gas tank	90
Oil tank	78
Electric tank	100

6. Select type of DWHR unit

Manufacturer	Model
Power Pipe	R3-60
Power Pipe	R3-36
GFX	G3-60
GFX	G3-40
Retherm	C3-40
Retherm	S3-60

7. Select closest city (will reference to cold water temperature table) – (COLDn, where n refers to month, 1-12))

8. Select configuration (CONFIG)

   A-Preheat cold water to hot water tank only
   B-Preheat cold water to hot water tank and shower

5.5.2. Other data required

1. Water supply temperature:

   This information is available in HOT2000 and could be supplied to web designer as lookup table.

2. NTU equations:

   The user input would be linked to a look-up table with the NTU-coefficients; $NTU = C \times HXF^{-n}$
   (HFX, heat exchanger flow)

Type	C	n
GFX-40	3.7669	0.6452
GFX-60	4.2096	0.6458
Ret-40	3.4053	0.7028
Ret-60	3.0710	0.5996
PP-36	2.8869	0.7219
PP-60	4.7622	0.6355

5.5.3. Calculation process

Data should be calculated on a monthly basis (n=1…12, and summed for an annual result, due to changes in cold water supply temperature.

User provides Ts, L, N, F, DHWe, DHWt, DWHR, COLDn, CONFIG

1- Determine HXF (heat exchanger flow) in L/min

If CONFIG=A,

HXF = F - [(F-(Ts x F/55))/(1-(COLDn/55))]

Else, HXF=F

2- Calculate NTU

Based on DWHR selection, lookup C and n and calculate NTU

$NTU = C \times HXF^{-n}$

3- Calculate effectiveness (ε)

If CONFIG=A

$$\varepsilon = \frac{1-\exp(-NTU(1-C_r))}{1-C_r\exp(-NTU(1-C_r))}$$, where C_r = HFX / F

Else,

$$\varepsilon = \frac{NTU}{1+NTU}$$

4- Calculate Heat Flux (q)

q = ε x (HFX/60) x 4.18 ((Ts-6)-COLDn)
6ºC loss from shower to drain

5 – Calculate Monthly savings (MSn, where n is month, 1-12) (kWh), corrected for hot water heater efficiency.

MSn = 0.216 x q x L x N x d / (DHWe) ,
d = number of days in given month
0.216 = 60/1000/3.6

6 – Calculate Annual savings (AS)

$AS = MS_1 + MS_2 + \ldots + MS_{12}$

7 – Convert to fuel type (CAS, converted annual savings)

If DHWt = 1 or 2 (natural gas, 37.3 MJ/m^3))

CAS = AS x 3.6 / 37.3 , result in m^3 ng

If DHWt= 3 (oil, 38.5 MJ/L)

CAS = AS x 3.6 / 38.5 , result in L oil

Else, CAS = AS (electric) kWh

6. Conclusion

The energy recovery and performance of each unit was measured and calculated using the different configurations. It was found that the tests could be standardized and that NTU correlations could be derived for each DWHR unit in order to develop an energy saving calculator. Pressure loss was also measured for each unit and correlation curves were plotted to predict the losses. A standard test setup was suggested which would permit the experiment to be easily repeated.

All tests were performed under relatively the same conditions, at the Canadian Centre for Housing Technology (CCHT). With the development of a standard testing method the experiments can easily be repeated in the future. It was found that the NTU test results were independent of the flow configurations and that the experiment could be greatly simplified by only varying the flow rate rather than the shower temperature, flow rate and configuration.

The best performing unit was the Power Pipe R3-60, and the best per foot result was the GFX G3-40 pipe. The unit with the overall least pressure loss was the Power Pipe 36. On a per foot basis both Power Pipe units performed equivalently well in terms of pressure loss. It was also found that there is an optimal balance between performance and size, as the pipes get longer they tend to only add marginal benefits to the performance; shorter pipes perform best on a per foot basis.

7. References

[1] Zaloum, Gusdorf, Parekh, ***Performance Evaluation of Drain Water Heat Recovery Technology at the Canadian Centre for Housing Technology, 2006***

[2] American Society of Heating, Refrigerating and Air-Conditioning Engineers, ***2005 ASHRAE Handbook Fundamentals***, pg 3-28

[3] Incropera, Dewitt, ***Introduction to Heat Transfer,*** John Wiley & Sons, 2002, pp. 606-644

[4] Kreith, Bohn, ***Principles of Heat Transfer***, Brooks/Coles 2001, pp.493-513

[5] Cengel, Boles, ***Thermodynamics An Engineering Approach***, McGraw-Hill, 2002

[6] University of Pittsburg,(2006), ***http://granular.che.pitt.edu/~mccarthy/che1011/Ex/Ex5/ex5.html***

8. Appendices

8.1. Appendix 1 : Fully detailed calculation

Empirical Calculation

Pipe:	GFX G3-60
Shower flow:	10.5 L/min
Cold water flow:	3.5 L/min
Hot water flow:	7.0 L/min
Configuration:	A
Drain water into pipe:	40.37ºC
Drain water out :	27.1 ºC
Cold water in:	8.2 ºC
Cold water out:	27.1 ºC

Date: July 21 2006

Effectiveness

$$\varepsilon = \frac{(T_{c,o} - T_{c,i})}{(T_{h,in} - T_{c,in})}$$

$$= \frac{(27.1 - 8.2)}{(40.37 - 8.2)}$$

$$= 0.5877$$

Specific Heat

$$C = \dot{m} c_p$$

An empirical curve was derived for the specific heat of water, although it only changes slightly in the calculation range, by the second decimal place which can be negligible between the maximum and minimum value

$$c_{p-cold} = 2E\text{-}09T^4 - 6E\text{-}07T^3 + 7E\text{-}05T^2 - 0.0028T + 4.2155$$

$$= 4.18 \text{ KJ/Kg C}$$

$$c_{p-hot} = 4.18 \text{ KJ/Kg C}$$

$C_{\min}$ = 4.18 (7.0/60) = 0.4882 KJ/s C

$C_{\max}$ = 0.731 KJ/s C

$$C_r = \frac{C_{\min}}{C_{\max}}$$

= 0.667

Heat Transfer

$$q = \varepsilon C_{\min}\left(T_{h,in} - T_{c,in}\right)$$

= 0.5877*0.4882*(40.36-8.2)

=9.23 KW

LMTD

$$\Delta T_{lm} = \frac{\left(T_{hi} - T_{co}\right) - \left(T_{ho} - T_{ci}\right)}{\ln\left[\frac{\left(T_{hi} - T_{co}\right)}{\left(T_{ho} - T_{ci}\right)}\right]} \quad \text{ºC}$$

$$= \frac{(40.36 - 27.11) - (28.3 - 8.2)}{\ln\left[\frac{(40.36 - 27.11)}{(28.3 - 8.2)}\right]}$$

= 16.44 ºC

Determine overall heat transfer coefficient from amount of heat transfer and LMTD

$$q = UAF\Delta T_{lm}$$

F =1 One pass counter flow heat exchanger

UA $= q / F\Delta T_{lm}$

=9.23/16.44

=0.561 KW/ºC

NTU method

$$NTU = \frac{1}{(C_r - 1)} \ln\left(\frac{\varepsilon - 1}{\varepsilon C_r - 1}\right) \qquad (C_{\min} \angle 1)$$

$$= \frac{1}{(0.667 - 1)} \ln\left(\frac{0.587 - 1}{0.587 * 0.667 \quad - 1}\right)$$

= 1.1668

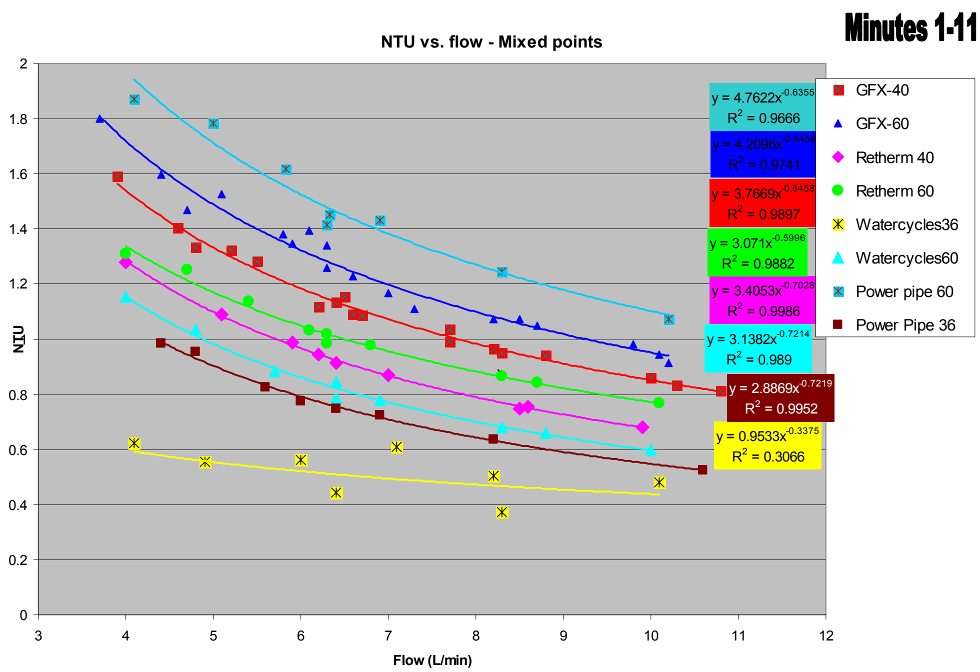
Determine overall heat transfer coefficient from the NTU

$$UA = NTU \cdot C_{\min}$$

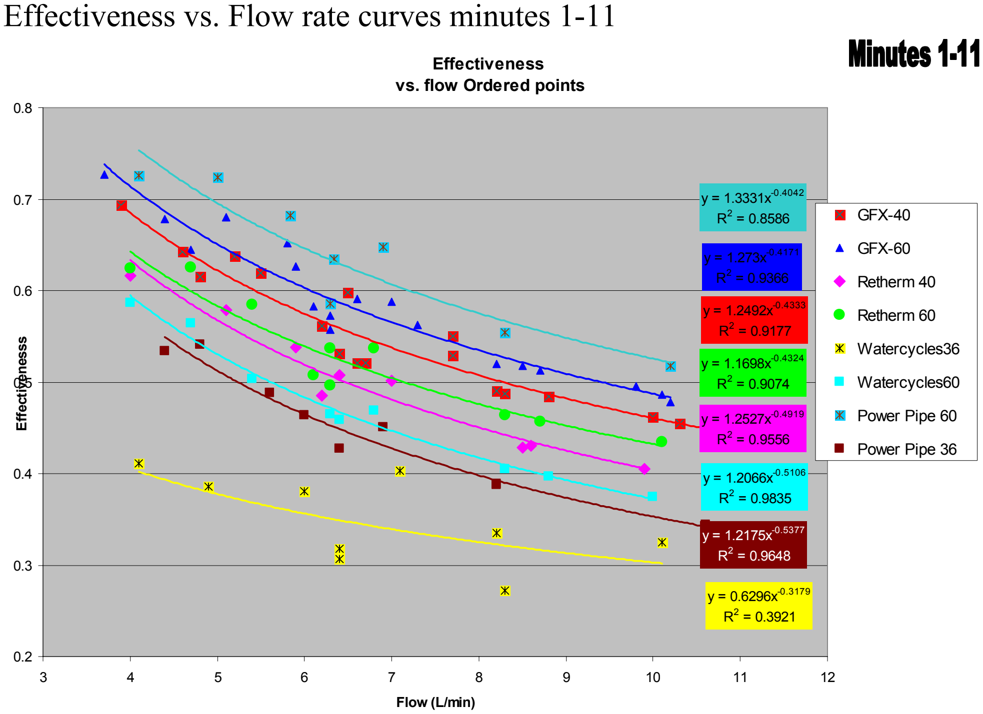
= 0.569 KW/ºC

8.2. *Appendix 2: NTU and Effectiveness Curves*

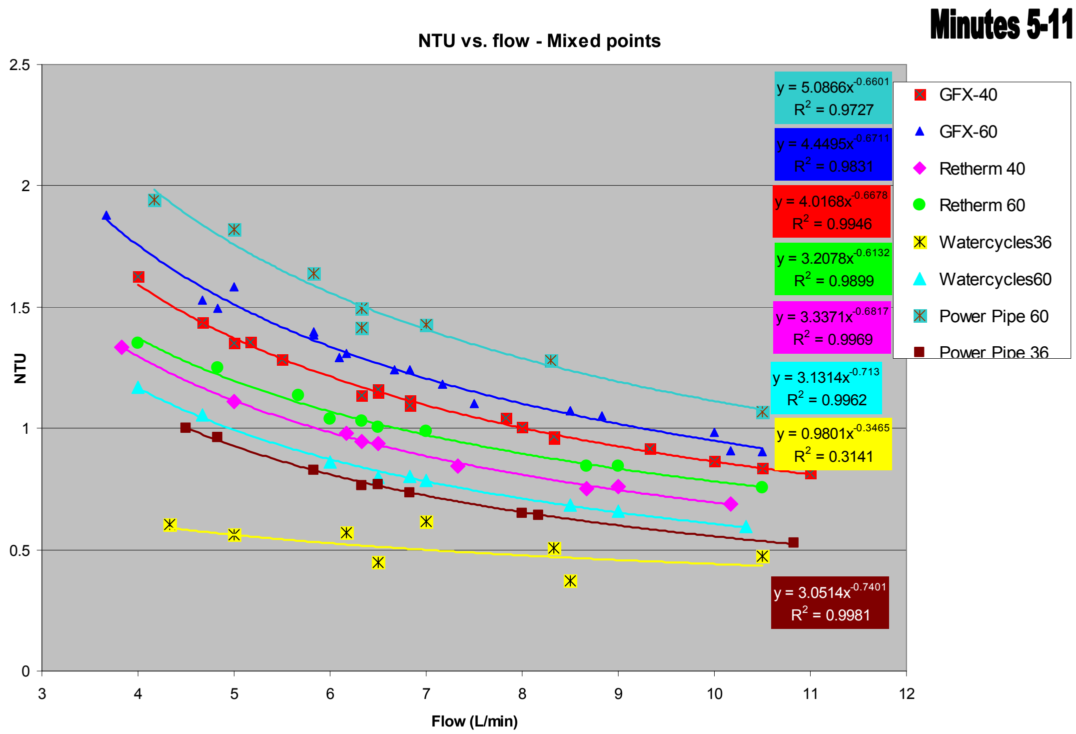
NTU vs. Flow rate curves minutes 1-11



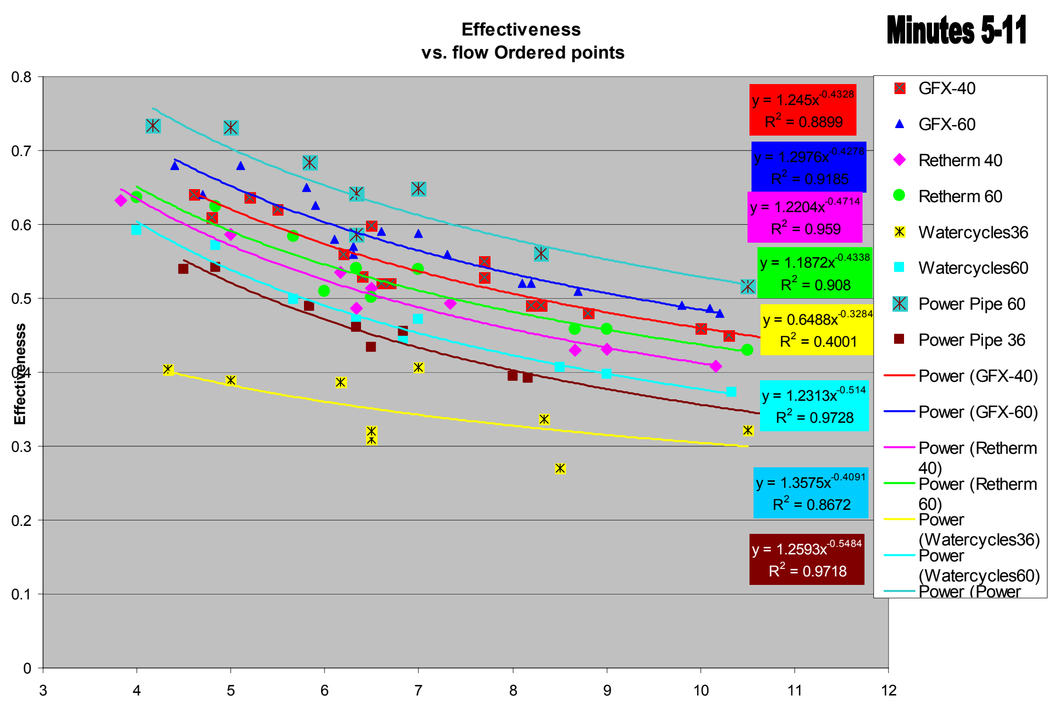
Effectiveness vs. Flow rate curves minutes 1-11



NTU vs. Flow rate curves minutes 5-11



Effectiveness vs. Flow rate curves minutes 1-11



8.3. Appendix 3: Example energy saving calculator and TRC analysis

DWHR model # from list	**6**	
Location	**Ottawa**	
Shower temperature	42	C
Hot water temperature	55	C
Hot water heater efficiency	89	%
Shower Flow	8.5	L/min
Ave. shower length	12	min
Shower frequency (per day)	4	
Configuration	B	(choice of A or B)

Model #	Type	C	n
1	GFX-40	3.7669	0.6452
2	GFX-60	4.2096	0.6458
3	Ret-40	3.4053	0.7028
4	Ret-60	3.0710	0.5996
5	PP-36	2.8869	0.7219
6	PP-60	4.7622	0.6355
7	WC-36	0.9533	0.3375
8	WC-60	3.1382	0.7214

Annual savings calculator

Month	Cold Water C	Cold Flow L/min	Hot Flow L/min	HX Flow L/min	Cr	NTU	E-NTU	q kW	Energy saved kWh/day	Energy saved kWh/mth	
Jan	7.5	2.33	6.17	8.50	1	1.22	0.55	9.93	7.95	276.80	
Feb	6.9	2.30	6.20	8.50	1	1.22	0.55	10.13	8.10	282.25	
Mar	7.5	2.33	6.17	8.50	1	1.22	0.55	9.93	7.95	276.80	
Apr	9.4	2.42	6.08	8.50	1	1.22	0.55	9.31	7.45	259.56	
May	11.9	2.56	5.94	8.50	1	1.22	0.55	8.50	6.80	236.87	
Jun	14.4	2.72	5.78	8.50	1	1.22	0.55	7.69	6.15	214.18	
Jul	16.3	2.86	5.64	8.50	1	1.22	0.55	7.07	5.65	196.94	
Aug	16.5	2.87	5.63	8.50	1	1.22	0.55	7.00	5.60	195.12	
Sep	16.3	2.86	5.64	8.50	1	1.22	0.55	7.07	5.65	196.94	
Oct	14.4	2.72	5.78	8.50	1	1.22	0.55	7.69	6.15	214.18	
Nov	11.9	2.56	5.94	8.50	1	1.22	0.55	8.50	6.80	236.87	
Dec	9.4	2.42	6.08	8.50	1	1.22	0.55	9.31	7.45	259.56	
									Annual	**3197.86**	**kWh**
									Converted to gas	**308.64**	**m3 gas**
									Fuel cost	**0.50**	**$/m3**
									Annual saving	**154.32**	**$**

Net Present Value Calculations for DWHR

Simplified calculation based on installed cost and saved energy only.

DWHR equipment life:	30	years.
Installed Cost:	$ 800.00	
Annual gas Savings:	308.64	m3 gas
Marginal Cost of Fuel:	0.50	$/m3 gas in Year 1
Fuel inflation rate:	2.5%	
Discount Rate:	6.0%	

Year	Cost of Elec ($/kWh)	Benefits	Costs	Net Benefit
1	0.500	$ 154.32	$ 800.00	-$ 645.68
2	0.513	$ 149.27	$ -	$ 149.27
3	0.526	$ 144.39	$ -	$ 144.39
4	0.539	$ 139.66	$ -	$ 139.66
5	0.553	$ 135.09	$ -	$ 135.09
6	0.567	$ 130.67	$ -	$ 130.67
7	0.581	$ 126.40	$ -	$ 126.40
8	0.596	$ 122.26	$ -	$ 122.26
9	0.611	$ 118.26	$ -	$ 118.26
10	0.626	$ 114.39	$ -	$ 114.39
11	0.642	$ 110.65	$ -	$ 110.65
12	0.658	$ 107.03	$ -	$ 107.03
13	0.675	$ 103.52	$ -	$ 103.52
14	0.692	$ 100.14	$ -	$ 100.14
15	0.710	$ 96.86	$ -	$ 96.86
16	0.727	$ 93.69	$ -	$ 93.69
17	0.746	$ 90.62	$ -	$ 90.62
18	0.765	$ 87.66	$ -	$ 87.66
19	0.784	$ 84.79	$ -	$ 84.79
20	0.804	$ 82.02	$ -	$ 82.02
21	0.824	$ 79.33	$ -	$ 79.33
22	0.845	$ 76.74	$ -	$ 76.74
23	0.867	$ 74.23	$ -	$ 74.23
24	0.889	$ 71.80	$ -	$ 71.80
25	0.911	$ 69.45	$ -	$ 69.45
26	0.934	$ 67.18	$ -	$ 67.18
27	0.958	$ 64.98	$ -	$ 64.98
28	0.982	$ 62.85	$ -	$ 62.85
29	1.007	$ 60.79	$ -	$ 60.79
30	1.032	$ 58.81	$ -	$ 58.81
Lifetime:		$ 2,977.83	$ 800.00	$ 2,177.83