Feasibility Study
of
Siphonic Turbine

Deltares

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Feasibility study of siphonic turbine

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Deltares

Title
Feasibility study of siphonic turbine

Client	Project	Reference	Pages
Provincie Zeeland, directie Ruimte, Milieu en Water	1208943-000	1208943-000-HYE-0002	21

Keywords
Siphonic turbine, Grevelingenmeer, Brouwersdam, Hydraulic performance, System efficiency.

Summary
The European project Pro-Tide aims to identify the Best Available Technology for conditions as encountered in the Grevelingen Brouwersdam-project. A preliminary multi-criteria analysis has ranked the Aerated Siphonic Turbine as second best available technology; the Fish-Friendly Turbine was ranked #1. The aerated siphonic turbine consists of a siphon with a restricted air-inflow at the crown of the siphon. Since Deltares has a vast experience in air pocket transport in inclined pipelines, which is the basis for the operation of siphonic turbine, Deltares has been asked to evaluate the performance of the siphonic turbine based on its knowledge and expert judgment.

Deltares has used available data from the CAPWAT project and other literature to examine the hydraulic behaviour of the siphon and to assess its performance with respect to efficiency and generated power. The following conclusions are made based on this assessment:

1. For the current configuration of the system, a blow-back flow regime with a high air head loss is expected.
2. The efficiency of the siphon depends mostly on the slope angle of the siphon legs, the pipeline diameter and up to a lesser extent the height of the air inlet.
3. In order to reach the maximum discharge expected in Grevelingen Brouwersdam-project, which is about 4500 m^3/s, the total number of 191 Inverted U tube siphonic turbines with a diameter of 3.2 m should be installed. This will lead to a maximum power generation of 3.14 MW.
4. The maximum efficiency of 7.2% is achieved for the inverted U tube siphon with a diameter of 1.3 m.
5. Decreasing the size of the siphon diameter from 3.2 m to 1.3 m slightly increases the efficiency (from 5.7% to 7.2%). However, with a smaller diameter, approximately 7.2 times the numbers of siphons should be installed in order to keep the total discharge through the structure constant.
6. The water levels of -1.25 and 0.0 (NAP +m) at the Noordzee and Grevelingenmeer respectively result in the highest efficiency for the system. However, due to a low absolute pressure in the crown of the siphon, there is a risk of water degassing which may hamper the efficiency for this water level difference. Therefore, the eventual efficiency may slightly be lower than the reported value.

References
Deltares proposal doc no. 1208943-000-HYE-0001, dated on 07/10/2013.
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1_{draft}	Jan 2014	A. Alidai		I. Pothof		B. van Vossen	
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Contents

1 Introduction

1.1 Background

The European Interreg NWE project Pro-Tide focuses on "Developing, Testing and Promoting Tidal Energy in coastal and estuarine zones". The project is led by the Province of Zeeland, with its partners the Province of South Holland and Rijkswaterstaat. Other partners are The Isle of Wight (UK), Port of Dover (UK), Laboratory Oceanography and Geoscience (LOG, FR) and Waterwegen en Zeekanalen (BE). The project focuses on further development of low velocity and low head Tidal Power Plants. Themes within the project are Technology, Ecology, Economy and Public Private Partnerships.

The Dutch project within Pro-Tide aims to identify the Best Available Technology for conditions as encountered in the Grevelingen Brouwersdam-project: Extremely low head (~ 1 m average) and large flow rates (~ 2500 m^3/s average). As similar conditions may be encountered in delta river applications, river power plants are also part of the considered applications. A preliminary multi-criteria analysis has ranked the Aerated Siphonic Turbine as second best available technology; the Fish-Friendly Turbine was ranked #1.

The aerated siphonic turbine consists of a siphon with a restricted air-inflow at the crown of the siphon, ref [4]. The air inflow is restricted by the turbine which generates power. The main advantage of this concept is that fishes do not pass the turbine blades. Therefore, this concept is fish-friendly, provided that the fish can withstand the short duration of negative pressure in the crown of the siphon.

Another advantage is that the siphon system is relatively economical to install. A water turbine will usually need to be mounted well below the lowest water level, so there is a large civil engineering cost involved in the installation. The siphon tube is relatively simple device, and it is mounted over the dam with only the intake and outlet below the water.

Moreover, the air turbines in siphonic system are small and inexpensive. They run fast, and may need no gearbox, whereas low-head water turbines are large and costly and they may need gearboxes to step up the rotation speed.

Last but not least, the air turbine can be placed onshore, or in any convenient place. Water turbines have to be installed in the water, obviously, and the generators must be coupled to them.

The performance of a prototype aerated siphonic turbine strongly depends on the two-phase flow in the downward sloping leg of the siphon. The Province of Zeeland has recognised that Deltares has acquired specific experimental and analytical expertise on two-phase flow in large-diameter downward sloping pipes in the Joint Industry Project CAPWAT (ref [5] and ref [6]). Therefore, Deltares has been asked to perform an analytical analysis of the siphonic turbine based on the applicable results obtained in the CAPWAT project.

1.2 Objective

The main objective of this study is to assess the hydraulic behaviour and the performance of the siphonic turbine with respect to generated power and efficiency.

1.3 Scope of work

The scope of work for this project is given as follows:

1. Analytical assessment of an aerated siphon with a geometry according to the Flakkeese Spuisluis (45º inclined legs),

2. Analytical assessment of a full scale aerated siphon, with optimal geometry (inverted U-tube, vertical legs).

In total four layouts are considered for this project. The layouts are different from each other in the angle of siphon legs, the diameter of the pipeline and the height of the air inlet.

The assessment determines the best layout and geometry for the current configurations of the siphon with respect to generated power and efficiency.

2 Applied data

The data used in this study are given in Table 2.1.

Table 2.1: Data applied in the current study.

Description	Symbol	Value	Comment
General info, ref [7]			
Atmospheric pressure	P_{atm}	101,325 Pa	
Seawater density	ρ_w	1027 kg/m^3	
Air density	ρ_{atm}	1.2 kg/m^3	20 °C and atmospheric pressure
Gravitational constant	g	9.81 m/s^2	
Heat capacity ratio or adiabatic index	γ	1.4	
Surface tension	σ	0.07 N/m	
Water levels, ref [2]			
Maximum water level Noordzee	HWL_N	+1.25 NAP +m[1]	
Minimum water level Noordzee	LWL_N	-1.25 NAP +m	
Maximum water level Grevelingenmeer	HWL_G	+0.3 NAP +m	
Minimum water level Grevelingenmeer	LWL_G	-0.3 NAP +m	
Siphon geometry, ref [1]			
Pipeline diameter[2]	D	3.2 m	a = b = 3.2 m
Height of crown	Z_{cr}	+ 6.150 NAP +m	
Height of air inlet	Z_{inlet}	NAP +m	To be determined
Slope length from air inlet to the top of horizontal section of pipe	L_{inlet}	m	To be determined
Total horizontal length of pipeline	L_{Hor}	45.35 m	
Wall roughness	k	1 mm	Assumed
Siphon hydraulic properties			
Air mass flow rate at air inlet	$\dot{m}_{air}$	kg/s	To be determined
Air pressure at air inlet	P_{in}	Pa	To be determined
Air density at air inlet	ρ_{in}	kg/m^3	To be determined
Turbine power	P_{tur}	W	To be determined
Hydraulic system efficiency	η	%	To be determined
Hydraulic constants, ref [8]			
Head loss coefficient of bending 45°	ξ_{45}	0.13	
Head loss coefficient of bending 90°	ξ_{90}	0.21	
Head loss coefficient of bending 180°	ξ_{180}	0.3	
Head loss exit/entrance	ξ_e	0.5	

1 *Normaal Amsterdams Peil (NAP) or Amsterdam Ordnance Datum is a vertical datum in use in large parts of Western Europe*

2 *The pipeline of Flakkeese Spuisluis has a square cross section. Here, the diameter means the hydraulic diameter of a cross section which is calculated as: $\boldsymbol{D_h} = 2(\boldsymbol{ab})/(\boldsymbol{a}+\boldsymbol{b})$, where **a** and **b** are the width and length of the rectangular cross section.*

3 System description

3.1 Siphonic turbine concept

The general configuration of the siphonic turbine (hereafter called system) is shown in Figure 3.1. The air inlet is located at the top of the siphon where the pressure is below atmospheric. This results in air flowing through the turbine and entering the siphon. The turbine is connected to a generator which produces electrical energy (the turbine and the generator are indicated with letters T and G in the figure).

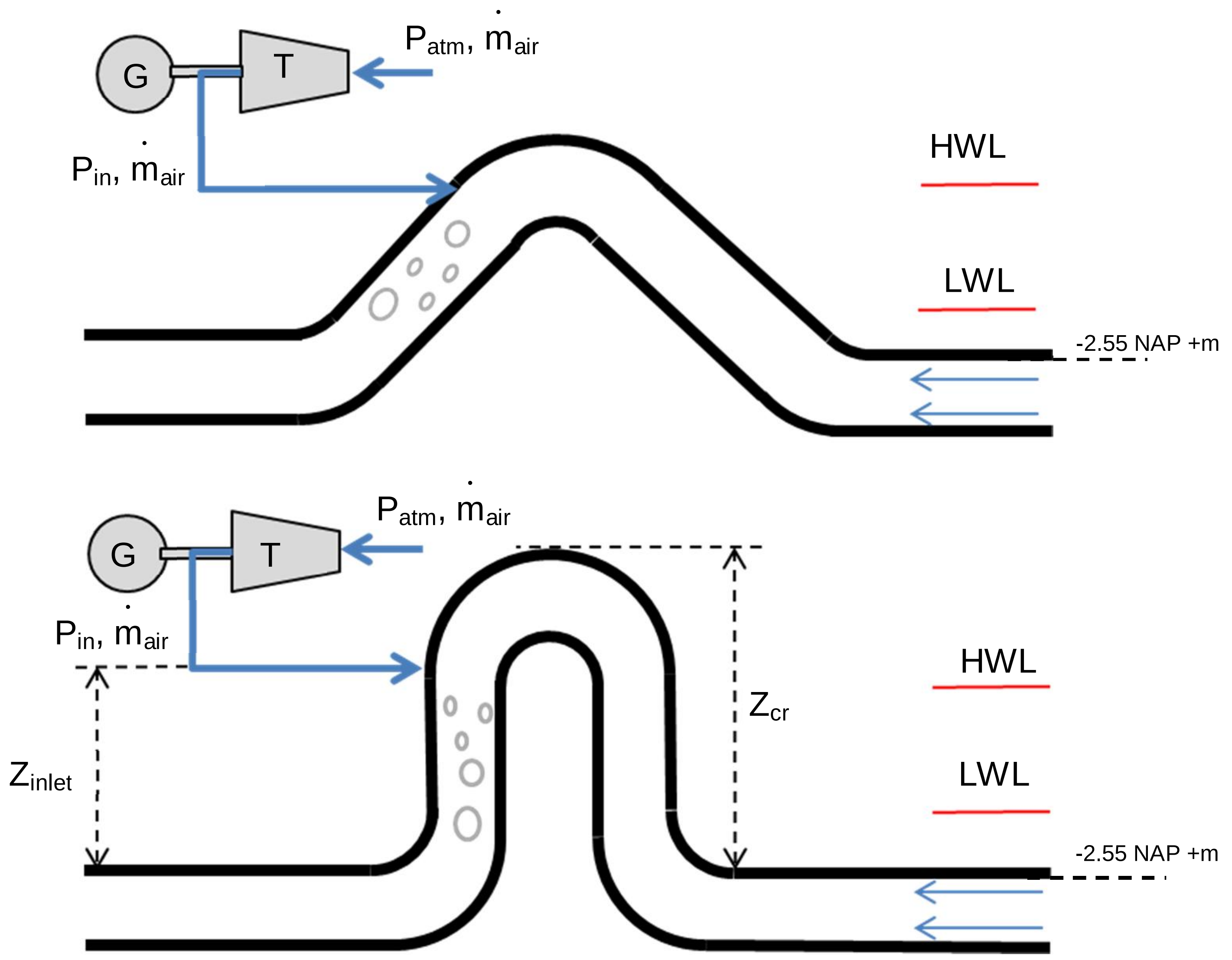

Figure 3.1: Schematic drawing of the system. Top figure demonstrates the layout according to the Flakkeese Spuisluis (45º inclined legs), and bottom figure depicts the inverted U tube siphon.

3.2 Relevant parameters

The relevant parameters, which influence the performance of the system, are divided into three categories:

1 Geometrical parameters; these parameters define the geometry of the system such as pipeline diameter, the height of the air inlet and the slope angle of the siphon legs. The pipeline diameter of 3.2 m is considered. Two slope angles of 45 and 90 degrees are examined in the current study. The optimum height of the air inlet is not known a priori and is determined based on the results of the study.

2 Boundary condition; the water levels upstream and downstream of the siphon. Figure 3.2 shows the schematised tidal water levels upstream and downstream of the siphon as a function of time, taking into account the envisaged exchange volumes with the

Grevelingenmeer, ref [2]. Here, two cases with maximum head difference are considered:
Case 1 Water level of -1.25 and 0.0 (NAP +m) at the Noordzee and Grevelingenmeer,
Case 2 Water level of +1.25 and 0.0 (NAP +m) at the Noordzee and Grevelingenmeer.

3 Flow rate of admitted air; the amount of the air admitted into the siphon influences the water capacity, the generated power, and the efficiency of the system. The optimum amount of the admitted air is determined based on the results of the study.

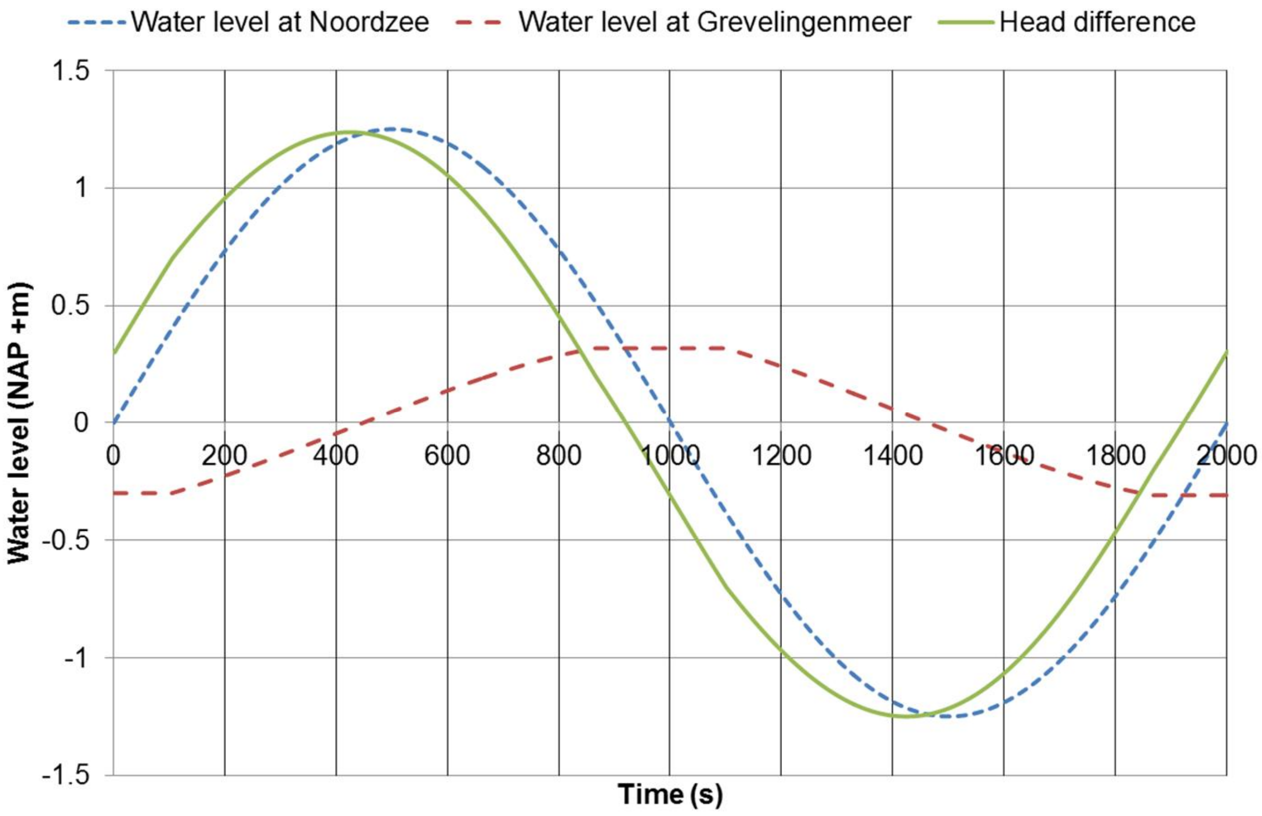

Figure 3.2: Schematised water levels upstream and downstream of the siphon as a function of time (ref [2]).

4 Approach and theoretical background

4.1 Steps

The following steps are taken in order to assess the performance of the system:

1. A 1-D hydrodynamic model of the system is built, taking into account friction losses and local losses such as bends and entrance/exit losses without considering air entrainment in the system. Using this model, the maximum water flow rate through the siphon is computed. This flow rate gives a good indication of the flow regimes which are expected in case of air admittance.

2. The maximum flow rate obtained in Step 1 is compared with the flow rates investigated in CAPWAT. If the values are in the same range, data reported in CAPWAT are directly used; otherwise the data are extrapolated from the reported values. The CAPWAT data provide a relation between the head loss caused by air pockets entrained in the system and the water flow rate. This relation is numerically included in the hydrodynamic model. Including all hydraulic losses (i.e. water head losses plus air pocket head losses) in the model, relevant hydraulic parameters to calculate the generated power and efficiency of the system are computed (more details of this computational procedure is given in the next section).

4.2 Theoretical background

In this section, a brief theoretical background about air transport in inclined pipelines is provided (for more details reference is made to CAPWAT where recent investigations on gas pocket transport are summarised). Moreover, required formulas to calculate the work done by the turbine and the efficiency of the system are outlined.

4.2.1 Flow regimes

Based on the discharges of the water and air, one of the following regimes occurs in an inclined pipe (ref [6] and ref [9]):

1. Stratified flow in entire slope. A single air pocket fills the entire slope, causing the maximum air pocket head loss. The air entraining hydraulic jump[3] occurs in the downstream horizontal section. The transition to the blow-back flow regime depends on the air flow number. The flow number of water and air (F_w and F_g) are defined as follows:

$$F_w = \frac{Q_w}{A\times(gD)^{0.5}} \;\; and \;\; F_g = \frac{Q_{air}}{A\times(gD)^{0.5}} \qquad \text{Equation1}$$

Where Q and A are the flow rate and the pipeline cross section.

2. Blow-back flow regimes 2a and 2b. The air pocket fills part of the slope with a water film underneath. The downward sloping pipe contains one or more air-entraining hydraulic jumps. The entrained air bubbles rise to the pipe soffit, coalesce to larger bubbles and secondary air pockets. These secondary air pockets have their own hydraulic jumps that

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3 *Hydraulic jump occurs when the liquid under the air pocket suddenly fills the pipeline. Liquid at high velocity discharges into a zone of lower velocity, a rather abrupt rise occurs in the liquid surface and air bubbles are ejected from the air pocket.*

pump the air further downward. The larger bubbles and air pockets blow back into the top air pocket. The larger the water flow rate, the smaller the upward velocity of the secondary air pockets. The air pocket head loss decreases gradually at increasing water discharge. Only a fraction of the entrained air reaches the bottom of the inclined section. The presence of a single air pocket and hydraulic jump is indicated by label 2a, the presence of multiple air pockets and hydraulic jumps by label 2b.

3 Plug flow regime. Stratified flow conditions and blow back phenomena do not occur anymore. A series of air plugs slowly moves in downward direction along the pipe soffit. The water hardly entrains air from the air pockets. The air pocket head loss has become marginal. The transition from flow regime 2 to 3 marks the dimensionless clearing velocity or required flow number, F_c, to keep elongated air pockets stationary. Figure 4.1 shows the clearing velocity for several slope angles.

4 Dispersed bubble flow regime. All air is transported in downward direction in dispersed bubbles and plugs. The air pocket head loss is negligible.

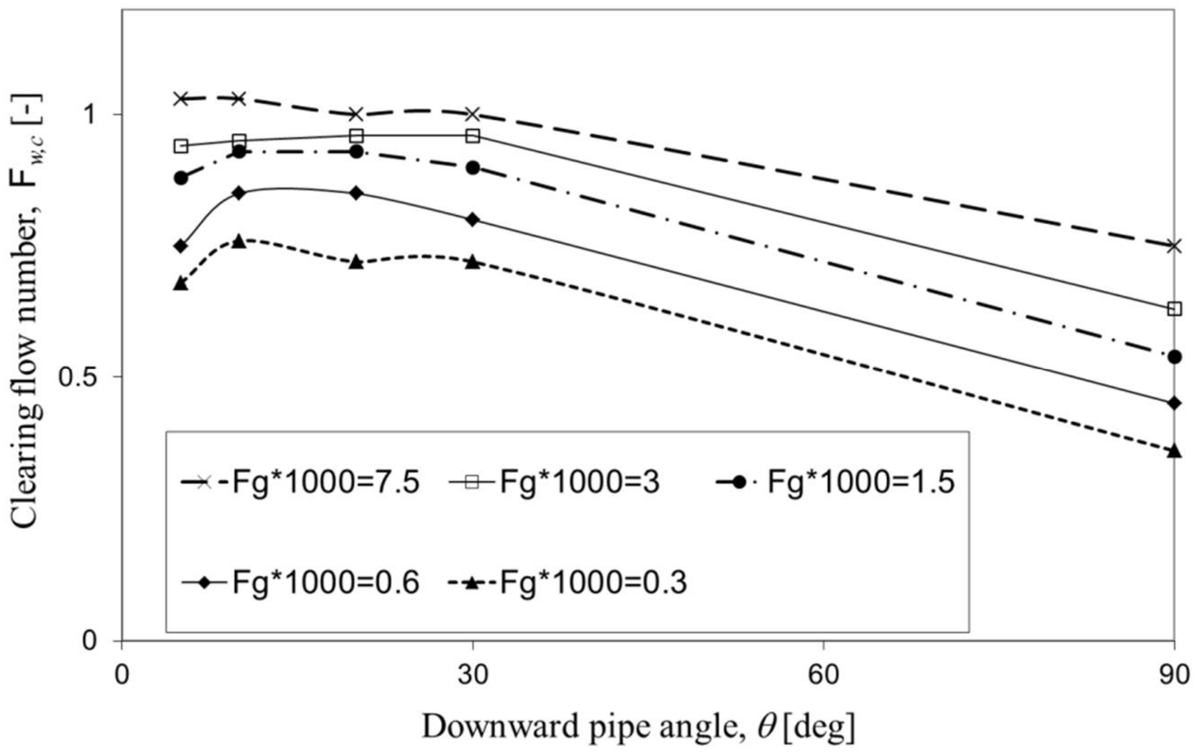


Figure 4.1: Clearing flow number as a function of slope angle (ref [6]).

4.2.2 Hydraulic losses

The total head loss in the system, which is equal to the available head imposed by boundary condition ($\Delta H_{total} = \Delta H_{avail}$), can be divided into: a) water head losses and b) air transport head loss (the first term and second term in Equation 2).

$$\Delta H_{total} = Constant \times v^2 + \Delta H_{air} \qquad \text{Equation 2}$$

The water head loss consists of the local losses such as bends and entrance/exit losses and the friction loss in the pipeline and is a quadratic function of the water velocity (ref [8]). There is not a straightforward formula to determine the head loss due to the air admittance and therefore, experimental data are used. In CAPWAT, it is shown that the air head loss in an inclined pipeline is a function of water and air flow rates. Figure 4.2 shows the experimental results for a vertical pipeline where the air head loss is determined for several admitted air flow numbers. For each air flow number, there is a relation between the air head loss and the water flow number (or simply the water velocity according to Equation 3):

$$v = F_w \times (gD)^{0.5} \qquad \text{Equation 3}$$

Using the experimental relation given in Figure 4.2 in combination with Equation 2, the water velocity and the air head loss can be calculated via an iterative numerical process.

It has to be noted that there is no data available in CAPWAT for slope angle of 45 degrees. Therefore, the data reported in another literature for a slope angle of 45 degrees are used (ref [10]) to obtain the relation between the air head loss and the water flow number. These data are compared with those of CAPWAT for a slope angle of 30 degrees to examine the consistency in the results of the two literatures.

Figure 4.3 shows the experimental data of both references. The air head loss in Kent's experiments is slightly lower in ref [10]. This is expected due to a number of differences:

1. A larger slope angle which enables an easier transport of air pockets in the pipeline,
2. A smaller pipe diameter (101 mm vs 220 mm), which promotes air transport. Scale effects are significant in pipes with diameters smaller than 200 mm,
3. An injection point in the downward sloping section, such that air accumulation in the top of the siphon is not possible anymore.

Nevertheless, the results are in good agreement with those of CAPWAT, and therefore, are used here for the layout with 45° siphon legs. For flow numbers less than 0.7 (F_w < 0.7), the data is linearly extrapolated to the point F_w = 0.1 and $\Delta H_{gas}/L\sin\theta$ = 1. This linear behaviour is commonly observed for this range of flow numbers (0.1 <F_w< 0.7) in this range of slope angles (ref [6]).

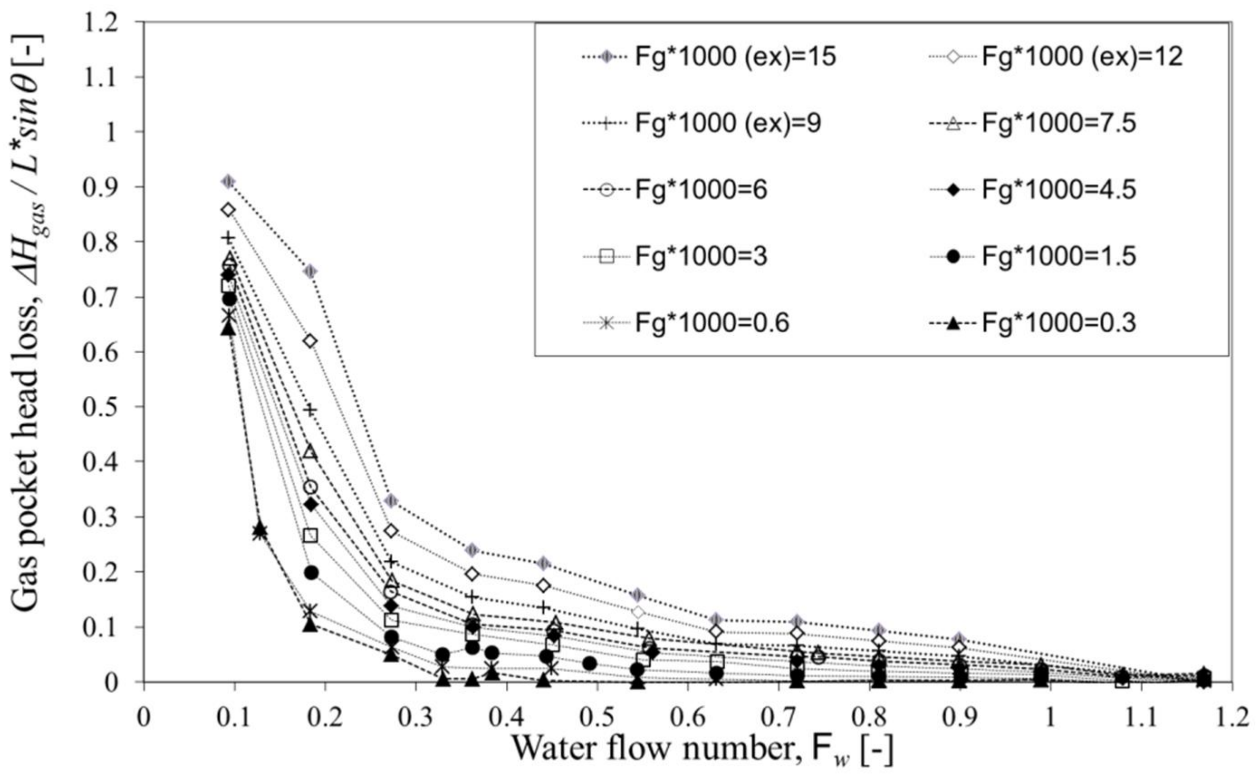


Figure 4.2: Air pocket head loss for a vertical pipe, D = 0.22 m, L/D = 10, 98 different air-water discharge combinations and 298 data points (ref [6]). L is the length of the pipeline, D is the pipeline diameter and θ is the slope angle of the pipeline. The data for the three highest air flow numbers are linearly extrapolated based on the smaller air flow numbers and are indicated with suffix "ex" in the legend.

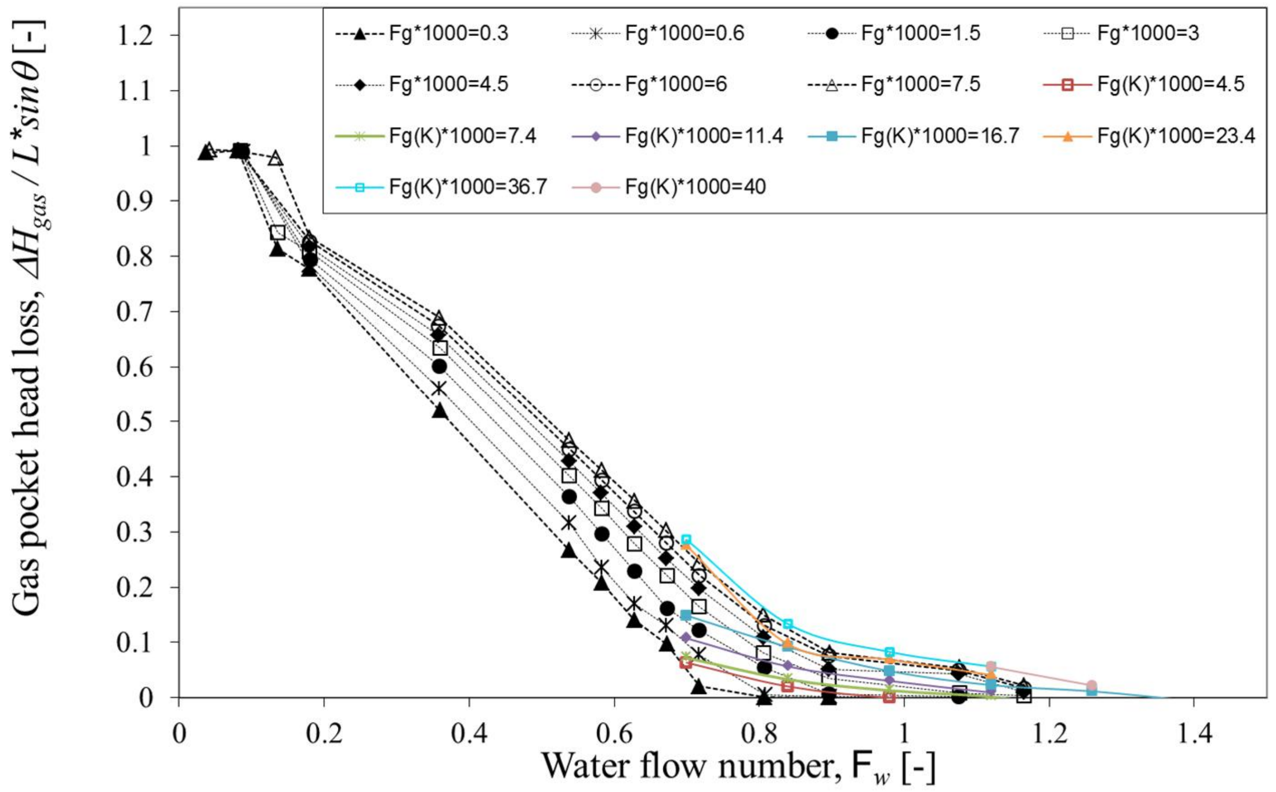


Figure 4.3: Black & white data show air pocket head loss obtained in ref [6] where θ = 30°, D = 0.22 m, L/D = 30. The coloured data are taken from ref [10] where D = 4 in, L/D = 48 and θ = 45°.

It has to be noted that an extensive investigation has been recently performed on the performance of the siphonic turbine (see ref [11]). However, the results of that study are not applicable here for the following reasons:

1 Due to larger water velocity (and consequently water flow number F_w), a bubbly flow regime was observed in ref [11] whereas for the current system a blow-back flow regime with large head loss is expected,
2 A larger available head and a shorter length of the sloping pipeline led to a considerably larger air intake (F_g) into the siphon.

4.2.3 Discussion on scaling effects due to slope length

The dimensionless geometry number, L_{inlet} /D, for the current project is about 1 to 2 depending on the height of the air inlet, Z_{inlet} (for definition of L_{inlet} and Z_{inlet} see Section 2). This is considerably lower than the ratio examined during CAPWAT project or the values given in other available literature (ref [9] and ref [10]). Nevertheless, similar flow regimes and hydraulic behaviour are expected in the current system. This is due to the fact the Eötvös[4] number is sufficiently high (Eo > 5000) to diminish the influence of diameter on the clearing flow velocity, but it is not completely clear how the head loss reduces at smaller slope length. Based on our expert judgment, the influence of the slope length on the head loss is not significant, and hence the available experimental results are exploited.

4.2.4 Air turbine performance and system efficiency

The details of the performance of the air turbine are out of the scope of this study. Therefore, following assumptions are made to obtain a simple relation between hydraulic parameters of the system and the performance of the air turbine:

[4] *Eötvös number (Eo) is regarded as the proportional of buoyancy force divided by surface tension force and can be calculated by* $Eo = \frac{\rho_w g D^2}{\sigma}$.

1 Air is considered as an ideal gas.
2 The work is done in the turbine through an isentropic expansion of the air. An isentropic process is a reversible adiabatic process which leads to highest generated power for a turbine. For a real turbine, the isentropic efficiency (proportion of real power to the isentropic power) ranges from 70-90%. Here, the losses in the turbine runner and rotational friction losses are ignored, implying that the efficiency of the turbine itself is idealised and set to 100%.
3 Local losses in air ducts are neglected.

Considering these assumptions, the generated power by turbine is calculated by Equation 4, ref [11]:

$$\frac{\mathrm{P_{turbine}}}{\dot{m}_{air}} = \left(\frac{\gamma}{\gamma-1}\right)\left(\frac{P_{atm}}{\rho_{atm}}\right)\left[\left(\frac{P_{in}}{P_{atm}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \qquad \text{Equation 4}$$

The system efficiency is defined as the work done by the turbine divided by the total input power:

$$\eta = \frac{\mathrm{P_{turbine}}}{\rho_w g \Delta H_{avail} Q_w} \qquad \text{Equation 5}$$

5 Results

5.1 Step 1: No air admitted

Table 5.1 shows the results obtained for two configurations of the siphon with no air admitted. The maximum water flow number is $F_{w,max}$ = 0.66. If air is admitted to the system, this number becomes even less since extra head loss will be introduced by air.

Figure 5.1 shows the clearing velocity required for several slope angles. Only for low air flow discharges (F_g×1000 < 4 in the figure), the clearing flow number can be reached in the current system. Therefore, for larger air discharges (F_g×1000 > 4), the blow back flow regime is expected which means a considerable air head loss.

Table 5.1 Results obtained for two configurations of the siphon with no air admitted.

Configuration	Available head H_{avai}(m)	Water velocity V_w (m/s)	Water discharge Q_w (m^3/s)	Water flow number F_w (-)
Flakkeese Spuisluis	1.25	3.68	37.75	0.66
inverted U tube	1.25	3.45	35.38	0.61

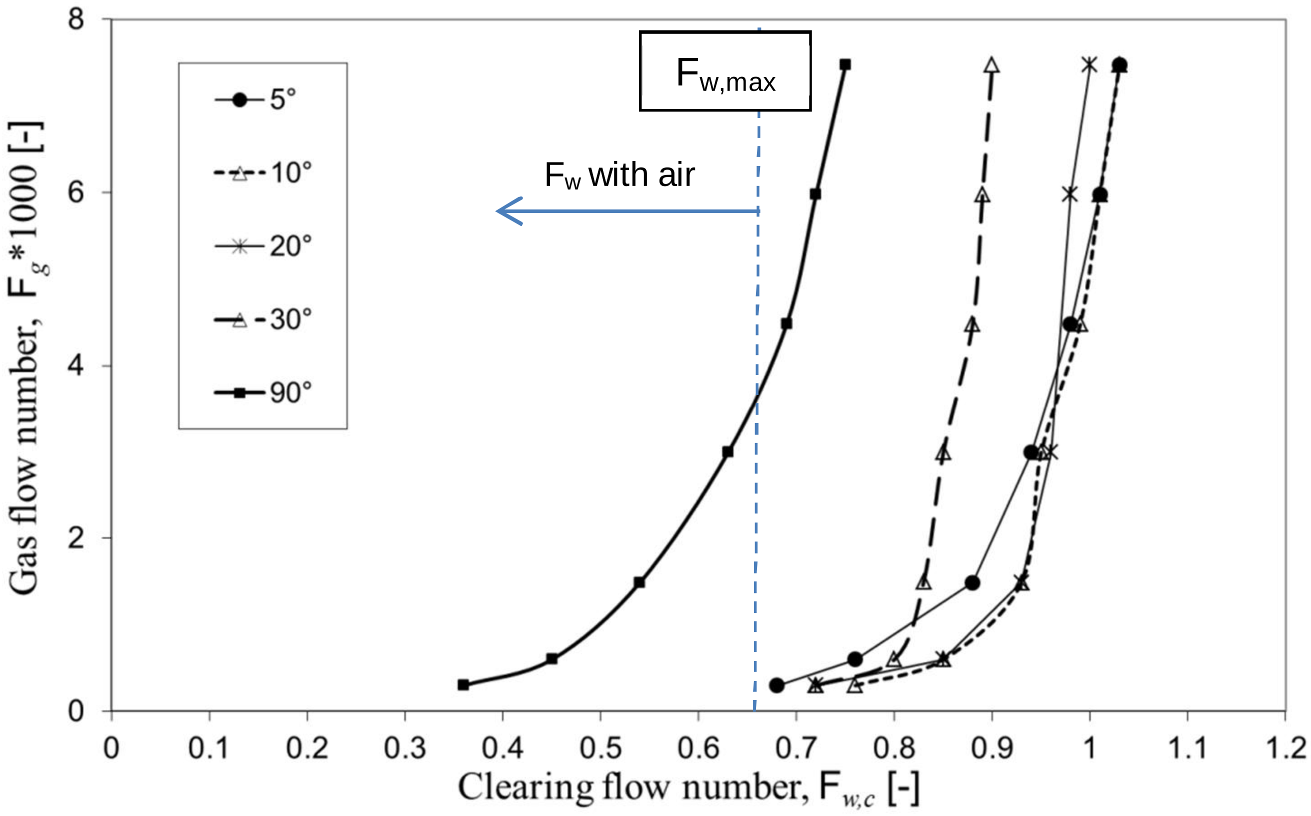


Figure 5.1: Clearing flow number as a function of admitted air obtained in CAPWAT project where slopes angle from 5 to 90 degrees are examined.

5.2 Step 2: Air admitted

5.2.1 Inverted U tube (Figures in Appendix A)

Figure 5.2 shows the water discharge for several air flow discharges as a function of the height of the air inlet. By increasing the admitted air (F_g), the head loss in the system increases, and therefore, less water can be transported. As it is seen in the figure, there is a limit to the admitted air. When F_g ×1000 reaches 15, the air head loss becomes so large that no water can be transported anymore and the siphon “breaks”. This is seen in Figure 5.3 by a sudden jump in the power and the efficiency to zero and the absolute pressure (at air inlet) to atmospheric pressure. In reality, when the air limit is reached, it takes some time for the air to fill the siphon leg and then pressure reaches atmospheric. Nevertheless, the exact flow behaviour in this situation is irrelevant for the current study where the steadily running siphon is the matter of interest.

The height of the air inlet also plays a role in the velocity of the water. The results show that by increasing the air inlet height, the water discharge decreases marginally (assuming F_g is constant).

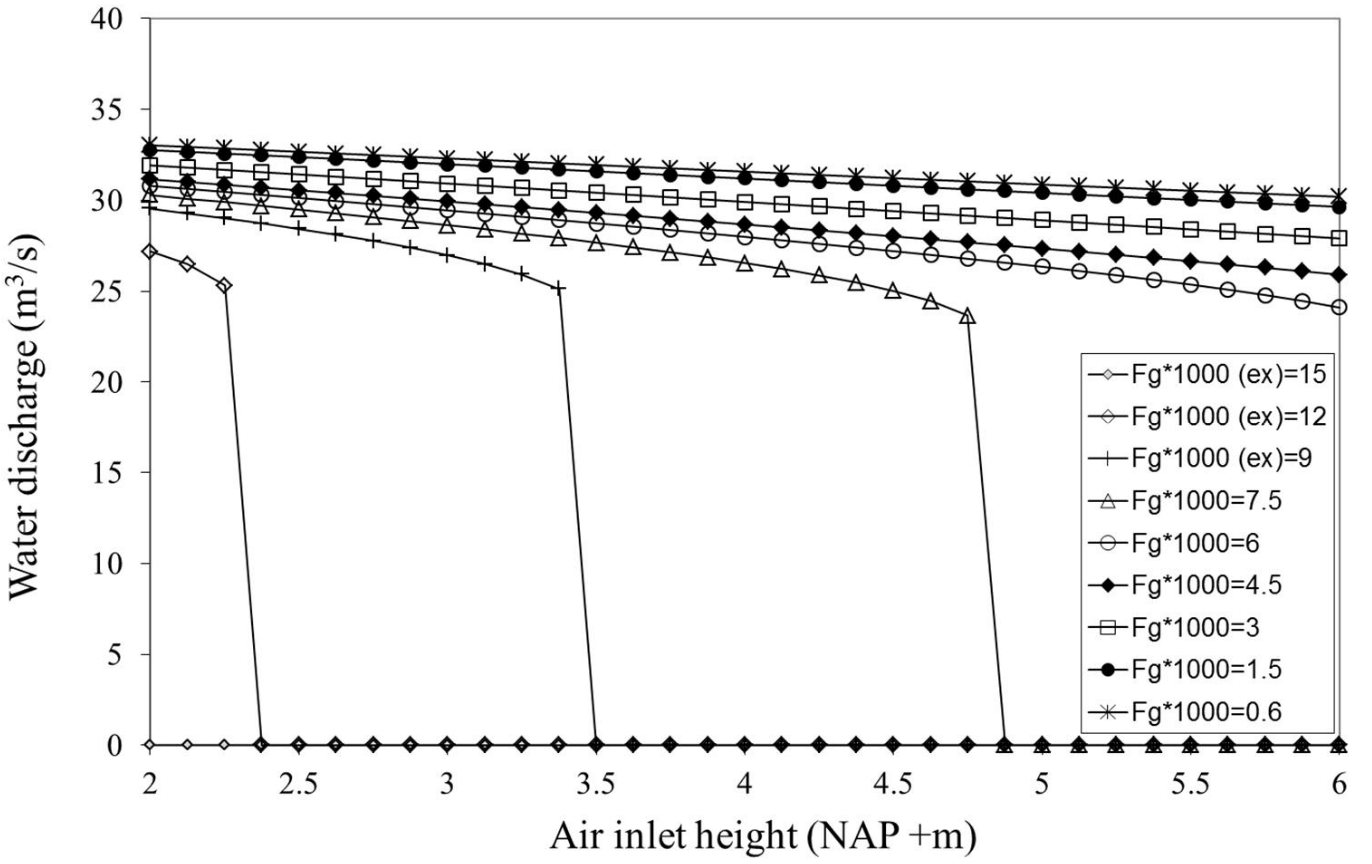


Figure 5.2: Water discharge as a function of the height of the air inlet for several admitted air discharges, $F_{g,}$ and the water level case 1 (see Section 3.2 for definition of water level case 1). F_g is the air flow number at the location of the air inlet.

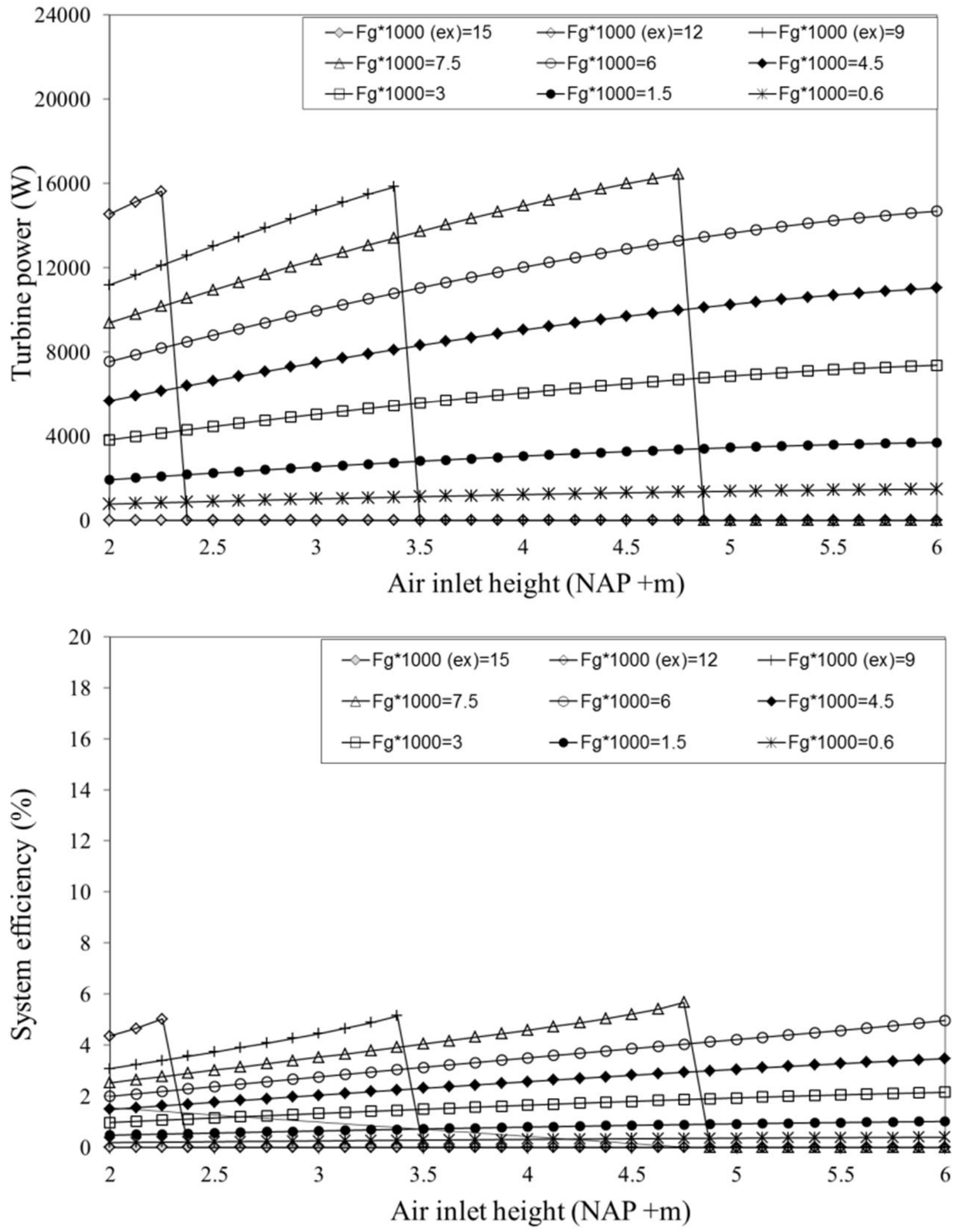
The power generated by the turbine and the efficiency of the system are directly related to the amount of air which is admitted to the system. As the air inlet elevation increases, both the differential pressure of the air turbine and the air flow rate increase, therefore, the generated power and system efficiency increase. As mentioned earlier, there is a limit to the increase of the admitted air due to air pocket blow back. The highest efficiency is about 5.7 % for the flow number $F_g \times 1000$ = 7.5 (corresponding to Q_{air} = 0.43 m^3/s at the air inlet) and the air inlet height of 4.75 NAP +m. The optimum design conditions with siphon pipes of 3.2 m diameter are listed in Table 5.2.

Table 5.2: Properties of the siphons with the highest efficiencies for inverted U tube siphon.

Case	Air inlet elevation (NAP +m)	Water flow rate (m^3/s)	Air flow rate (m^3/s)	Air pocket head loss (m)	Air inlet pressure (bar abs.)	Power (kW)	Efficiency (%)
1, low tide	4.75	23.6	0.43	0.69	0.52	16.4	5.7
2, high tide	4.75	23.6	0.43	0.69	0.64	13.4	4.60

The expected two-phase flow regime is the blow-back flow regime, because the flow number at these conditions is 0.52. Therefore, the air pocket head loss is 0.69 m, which is 55 % of the available head of 1.25 m. The height of the air inlet also affects the efficiency and the generated power. This is due to the fact that at higher locations in the siphon, the local absolute pressure is lower, and therefore, air expands to a lower pressure in the turbine. Consequently, more power is generated. This effect explains the difference in the results obtained for water level case 1 and case 2 where the head difference is equal but the absolute pressure in the siphon is different (compare figures in Appendix A.1 and Appendix A.2).

It has to be noted that at low pressures (P < -0.5 barg) the air bubbles inside the water start dissolving and form larger air pockets which increase the air head loss and hamper the efficiency of the system. This means that the real efficiency is slightly lower for the situations when the pressure drops below this value. However, quantification of the influence of this phenomenon is not straightforward and is not considered in the current investigation.



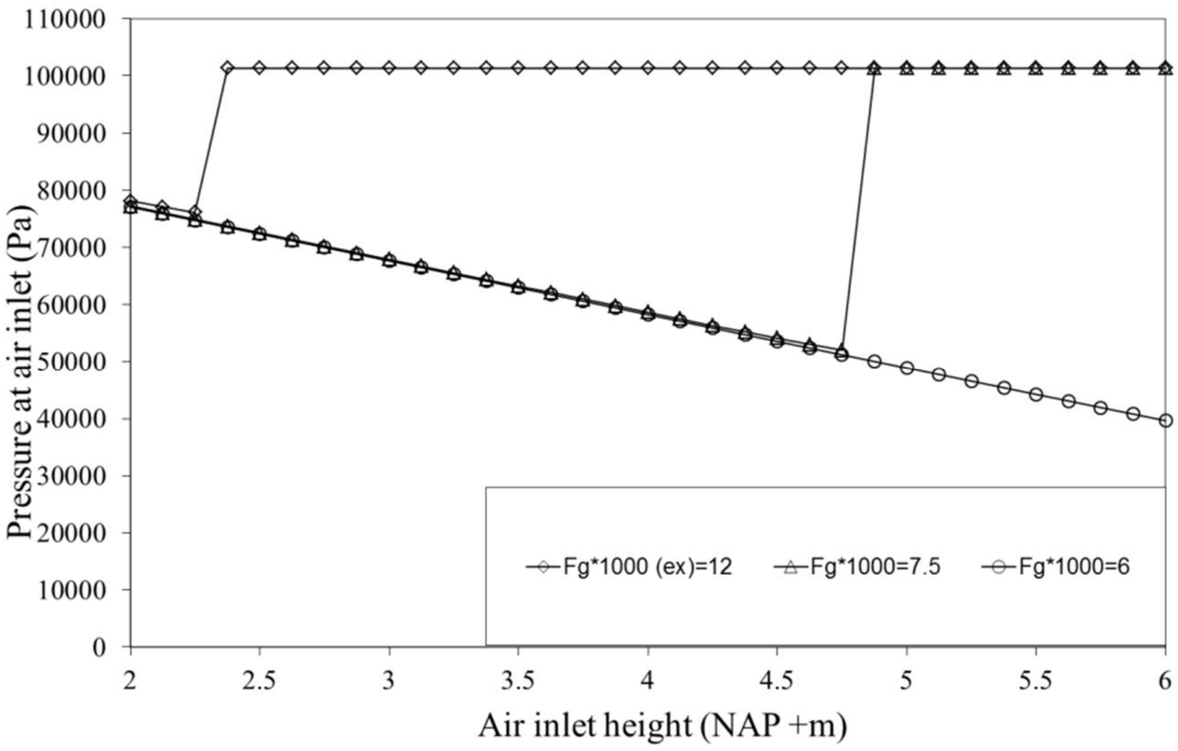


Figure 5.3: Generated power per siphon pipe, system efficiency and the absolute air pressure (at the air inlet) as a function of the air inlet height for several air discharges and water level case 1 (see Section 3.2). For better clarity in the bottom figure, only the absolute pressure for three air flow numbers is demonstrated.

5.2.2 Flakkeese Spuisluis (Figures in Appendix B)

For a range of air discharges for which the experimental results are available (i.e. $4.5 < F_g \times 1000 < 36.7$ shown in Figure 4.3), no water can be transported through the siphon, and therefore, no power is generated. This is due to a significant air head loss caused by the buoyancy force which moves the air pockets in opposite direction of main flow. It might be possible to admit a smaller amount of air in order to run the system. However, this means a frivolous power production and minute efficiency.

In order to reduce the air head loss, one solution is to reduce the size of the siphon diameter. By doing so, the water flow number F_w increases, and consequently, the air pockets can be transported with less head loss. This is due to the fact that at higher water velocities, the drag forces between water and air become dominant over the buoyancy forces on the air pockets.

Figure 5.4 shows the generated power and the system efficiency for the siphon diameter of 2 m. For this diameter, the maximum air flow number of $F_g \times 1000 = 7.4$ is feasible. This amount of air leads to a maximum system efficiency of about 3.2 %.

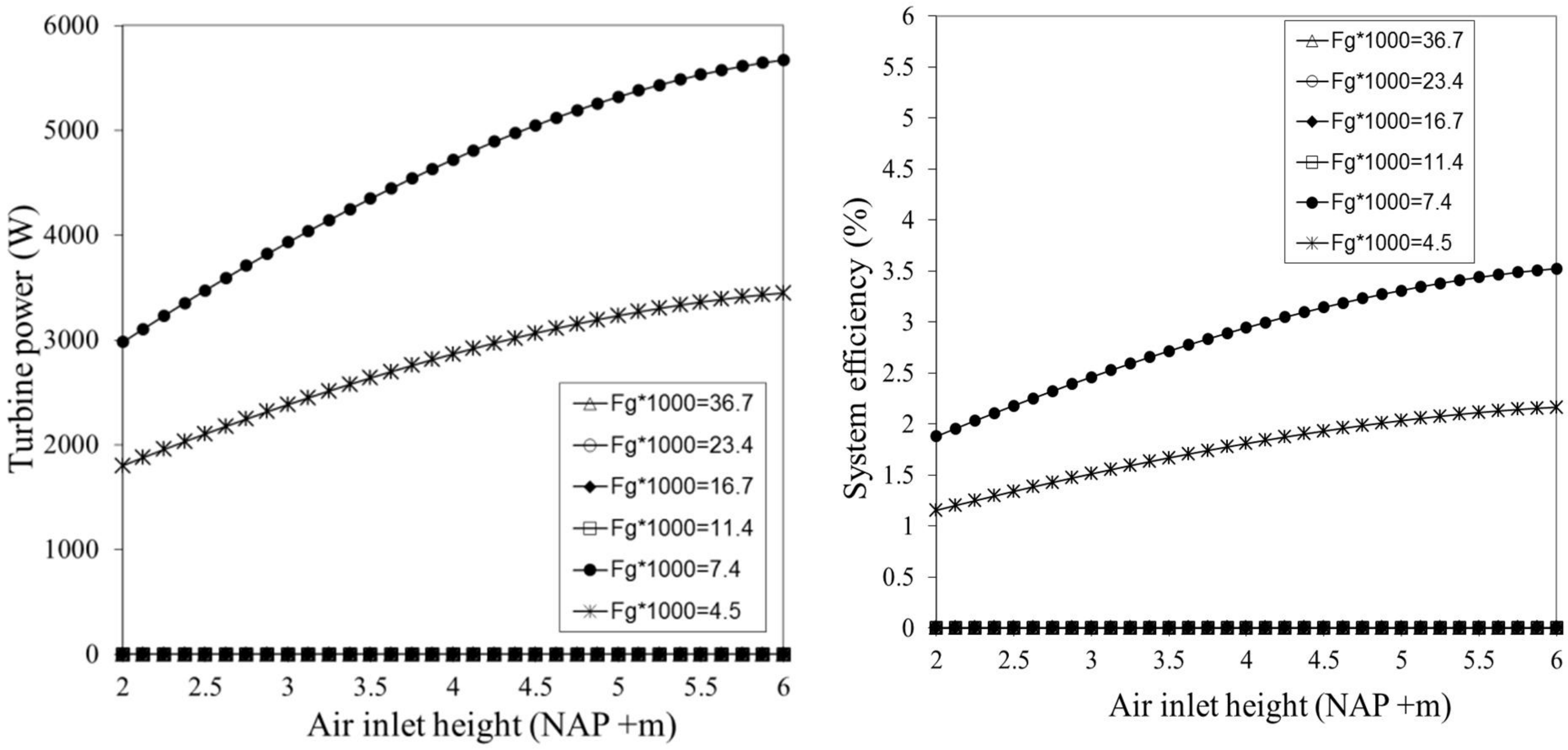


Figure 5.4: Power generated by the air turbine and system efficiency as a function of the air inlet height for several air discharges and water level case 1.

5.2.3 Optimum design (Figures in Appendix C)

Based on the previous sub-sections, it is concluded that both the slope angle of the siphon legs and its pipeline diameter influence the performance of the system. Deltares is requested to optimise the siphon configuration with respect to the efficiency regardless of cost considerations for large number of required siphons in case of small pipeline diameters (ref [3]). Therefore, a sensitivity analysis is performed where three parameters are changed namely, amount of air intake, air inlet elevation, and siphon diameter. Pipe diameters from 1 m to 3.2 m are checked. Based on the sensitivity analysis shown in Figure 5.5, the highest efficiency of 7.2 % is observed for a pipe diameter of 1.3 m (see Table 5.3).

Table 5.3: Properties of the siphons with the highest efficiencies achievable in the system.

Case	Air inlet elevation (NAP +m)	Water flow rate (m^3/s)	Air flow rate (m^3/s)	Air pocket head loss (m)	Air inlet pressure (bar abs.)	Power (kW)	Efficiency (%)
1, low tide	3.62	3.26	0.09	0.73	0.63	2.9	7.2

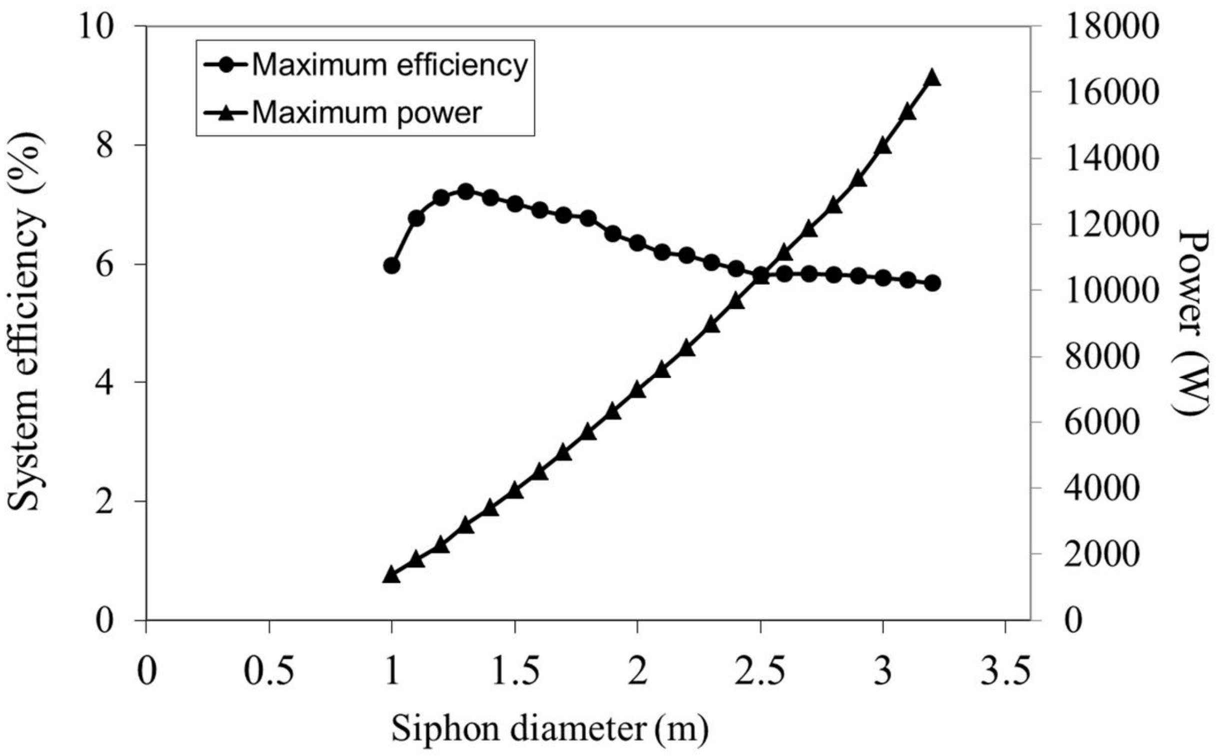


Figure 5.5: Maximum system efficiency and power per pipe as a function of pipeline diameter. The air inlet height and amount of admitted air may differ in each point.

The change in the diameter has two main implications for the efficiency which counteract each other, and therefore, the efficiency does not strongly depend on diameter. By decreasing the diameter, the water flow number, F_w, increases. This means that less air head loss in the system occurs. At the same time, the water head loss in the system increases due to the smaller pipe diameter and Reynolds number which means greater friction factor in the pipeline. Therefore, the water velocity will be reduced through the siphon. Lower velocities result in less air entrainment and consequently less power production. According to Equation 4 and 5, efficiency is related to the entrained air and water discharge by $\eta \sim \frac{\dot{m}_{air}}{Q_w}$. In case of smaller diameter, both nominator and denominator decrease. Hence the efficiency only changes slightly.

5.2.4 Discussion on results

The presented results in the previous sub-sections show a large discrepancy in efficiency with the reported values in ref [12]. In order to have a better understanding of the reasons for this discrepancy, a quantitative comparison between the results is provided in the following.

Figure 5.6 shows the head losses for the current system and the ones reported in ref [12]. As it is seen in the figure, the total head loss (or available head) in the ref [12] is almost twice as the available head in the current study. This has two implications:

1. The water flow number is significantly greater in ref [12] (about F_w = 1.5 when F_g = 0, see Table 5.4 for water and air flow numbers in both studies). This means that for an equal air flow number, the air head loss in ref [12] is much lower or for an equal amount of air head loss, much larger air flow numbers can be achieved in ref [12]. The relation between the air head loss and water flow number was discussed in Section 4.2.2. As demonstrated in Figure 4.2, by increasing the water flow number, the air head loss dramatically decreases, at a given air intake. For smaller water flow numbers, the air head loss is considerably larger especially for large amount of air intake. For example, the air head loss is almost 4 times more at F_w = 0.5 compare to that at F_w = 1.0 for the air flow number of F_g*1000 = 15.

2 Due to a larger available head, a greater air flow number can be reached even if the water flow number is the same. This is simply due to the fact that a larger driving force is available in the system to overcome the head losses.

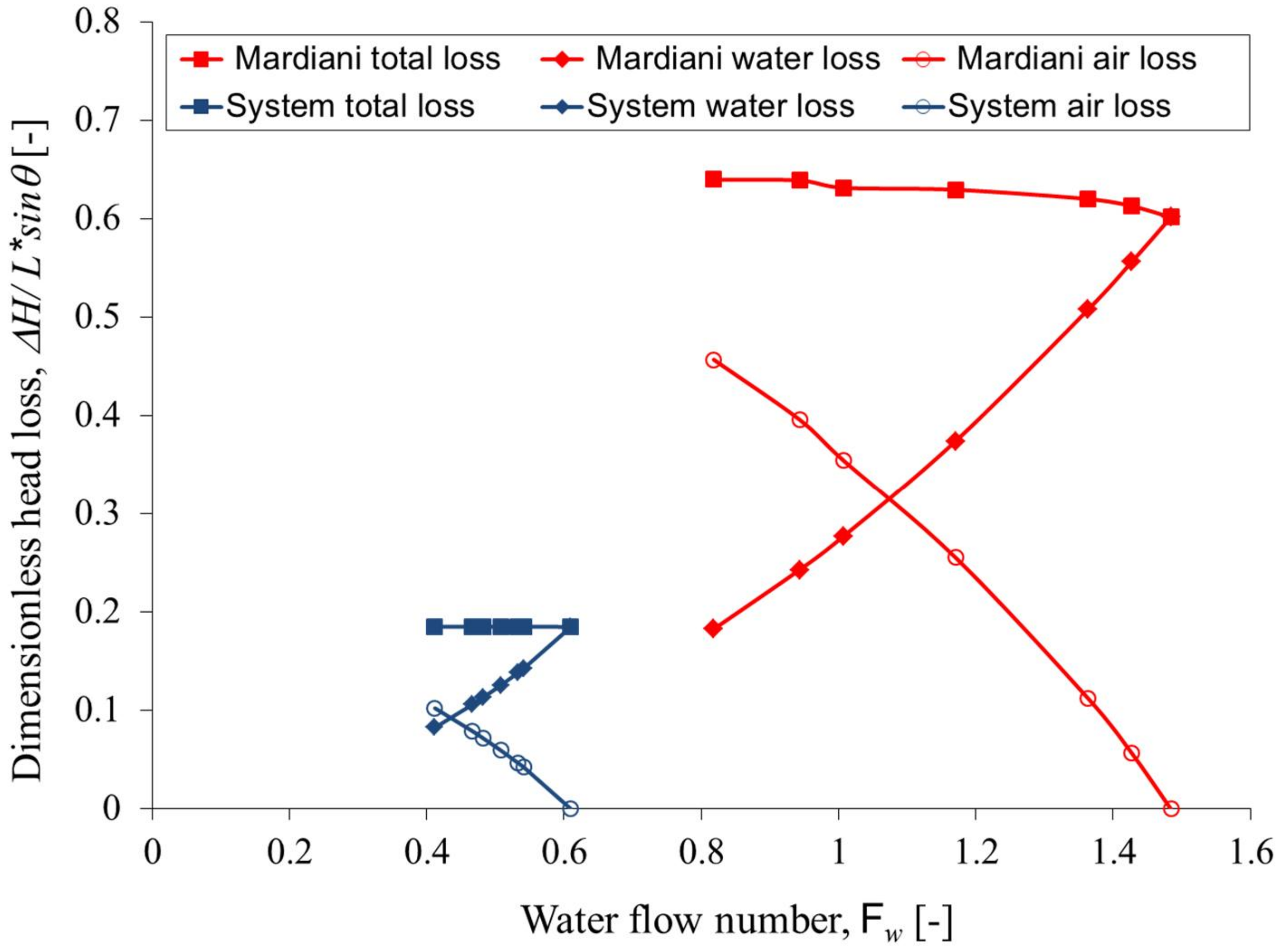


Figure 5.6: Total head loss, water head loss and air head loss. Blue curves show data in the current study for an inverted U tube siphon with D= 3.2 m and Z_{inlet} = 4.75 NAP +m and red curves are based on data described in ref [12] for an inverted U tube siphon with D = 0.2 m and L_{inlet} = 3.5 m.

Therefore, a larger available head results in a greater air-water ratio in ref [12] (see Table 5.4). With respect to the efficiency of the systems, a larger air-water ratio leads to a higher efficiency. Using Equation 4 and 5 (described in Section 4.2.4) the ratio of the efficiencies of the systems is calculated by Equation 6. Subscripts “s” and “m” stand for the current system and the one studied by Mardiani (ref [12]).

$$\frac{\eta_s}{\eta_m} = \frac{P_{turbine,S}}{\rho_{w,s} g \Delta H_{avail,s} Q_{w,s}} \frac{\rho_{w,m} g \Delta H_{avail,m} Q_{w,m}}{P_{turbine,m}} = \frac{\dot{m}_{air,s}}{Q_{w,s}} \frac{Q_{w,m}}{\dot{m}_{air,m}} \frac{\Delta H_{avail,m}}{\Delta H_{avail,s}} \frac{\varphi_{air,s}}{\varphi_{air,m}} \quad \text{Equation 6}$$

Where $\varphi = \left(\frac{\gamma}{\gamma-1}\right)\left(\frac{P_{atm}}{\rho_{atm}}\right)\left[\left(\frac{P_{in}}{P_{atm}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$.

The contribution of each term in Equation 6 to the efficiency ratio is calculated for two cases in the two systems (numbers in red and blue fonts in Table 5.4).

$\frac{\dot{m}_{air,s}}{Q_{w,s}} \sim 0.014$ and $\frac{\dot{m}_{air,m}}{Q_{w,m}} \sim 0.32 \rightarrow \frac{\dot{m}_{air,s}}{Q_{w,s}} \frac{Q_{w,m}}{\dot{m}_{air,m}} \sim 0.044$ 1- Air-water ratio contribution

$\frac{\Delta H_{avail,m}}{\Delta H_{avail,s}} \sim 1.8$ 2- Available head contribution

$\frac{\varphi_{air,s}}{\varphi_{air,m}} \sim 3.0$ 3- Pressure contribution

$\frac{\eta_s}{\eta_m}$ ~0.044 × 1.8 × 3.0~ 0.24 4- Efficiency ratio

Hence, the circumstances of Mardiani's set-up lead to an efficiency which is 4.2 times higher than the maximum efficiency at our pilot site. The most contribution to the efficiency ratio is by the air-water ratio. As explained earlier, larger available head results in a considerably greater air-water, and consequently higher efficiency ratio in ref [12].

Table 5.4: Results obtained in the current study and in ref [12]. The indicated data correspond to the data point shown in Figure 5.6.

Current system						
Water flow number F_w (-)	**Air flow number F_g*1000 (-)**	**Water discharge Q_w (m^3/s)**	**Air discharge Q_g (l/s)**	**Air inlet pressure (bar abs.)**	**Air mass flow rate $\dot{m}_{air}$ (kg/s)**	**Efficiency (%)**
0.61	0.0	35.4	0	48596	0.000	0.0
0.54	0.6	31.0	34.424	50008	0.025	0.4
0.53	1.5	30.6	86.06	50140	0.062	0.9
0.51	3.0	29.1	172.12	50564	0.126	1.9
0.48	4.5	27.7	258.18	50972	0.190	2.9
0.47	6.0	26.8	344.24	51211	0.254	4.0
0.41	**7.5**	**23.6**	**430.3**	**51986**	**0.321**	**5.7**
Mardiani ref [12]						
1.48	0.0	0.065	0	72375	0.000	0.0
1.43	124.3	0.063	5.47	73960	0.007	9.0
1.36	163.6	0.060	7.197	75056	0.009	12.0
1.17	201.6	0.052	8.871	76184	0.011	16.0
1.01	205.9	0.044	9.061	78547	0.011	19.0
0.94	205.5	0.042	9.042	82378	0.011	20.0
0.82	**219.9**	**0.036**	**9.678**	**83053**	**0.012**	**24.0**

6 Conclusions and recommendations

This section describes the conclusions and recommendations of the hydraulic assessment of the siphonic turbine.

6.1 Conclusions

1 For the current configuration of the system, a blow-back flow regime with large air head loss in the siphon is expected. Much smaller head loss occurs in systems with larger head difference where the water velocity is sufficiency large to disintegrate the air pockets into small bubbles moving in the flow direction.

2 The efficiency of the siphon depends mostly on the slope angle of the siphon leg, the pipeline diameter of the siphon and up to a less extent on height of the air inlet.

3 Highest efficiency is obtained for a siphon with the following properties:

Table 6.1: Properties of the siphon with the highest efficiency.

Best option	Efficiency (%)	Power (kW)	Water flow rate (m^3/s)	Configuration	Diameter (m)	Air inlet elevation (NAP +m)
	7.2	2.9	3.26	Inverted U tube	1.3	3.62

4 Decreasing the size of the siphon diameter increases the efficiency slightly (from 5.7% to 7.2%). However, with a smaller diameter, approximately 7.2 times the numbers of siphons should be installed in order to keep the total discharge through the structure constant.

5 The water levels of -1.25 and 0.0 (NAP +m) at the Noordzee and Grevelingenmeer respectively result in the highest efficiency for the system. However, due to a low absolute pressure in the crown of the siphon, there is a risk of water degassing which may hamper the efficiency for this water level difference. Therefore, the eventual efficiency may slightly be lower than the reported value.

6 In order to reach the maximum discharge expected in Grevelingen Brouwersdam-project, which is about 4500 m^3/s (ref [2]), the total number of 1381 siphonic turbines (with properties given in Table 6.1) should be installed. This will lead to maximum power generation of 4.0 MW.

7 For the current system the available submergence, which is the vertical distance from the water surface to the centreline of the horizontal pipeline, is rather small (about 4.15 m for water level case 1 and 5.35 m for case 2). This is almost half of the recommended value for the submergence of the intake to avoid air intrusion for a siphon diameter of 3.2 m (ref [13]). Therefore, there is a chance that free surface vortices form at the entrance of the pipeline which may draw air into the pipeline. This will lead to hampering of the efficiency and performance of the siphonic turbine. For a siphon diameter of 1.3 m, the submergence is sufficient and the free surface vortices are unlikely to occur.

8 It was shown that the amount of the transported air (and consequently the performance of the siphonic turbine) strongly depends on the available head in the system. For the current system, the available head of 1.25 m is relatively low and therefore, only a small

amount of air can be transported through the siphon. For sites with sufficiently large available head (> 2 m), a better performance for the siphonic turbine is expected especially in case of the rivers, where the available head is constant. A more detailed investigation should be performed to determine the maximum achievable efficiency of the sites with available head of more than 2 m.

6.2 Recommendations

The current study gives an indication of the expected performance of the siphonic turbine. However, there are several uncertainties which may lead to deviation of results in a real system, and hence should be addressed in a more detailed analysis or even experimental tests:

1 Length of downward sloping section: there is no experimental data available for a short length of inclined pipeline. A physical model test can be performed to obtain air discharge-head loss relations for such a short length.
2 Actual air turbine efficiency: the real efficiency depends on the type of the turbine which will be used. This efficiency can be provided by the turbine manufacturer.

7 Reference

7.1 Reference document and drawing

1 Drawing “Flakkeese spuisluis - Grevelingendam-Hevel”. Document no. GB. 604. Sent by email dated on 26-11-2013.

2 Drawing “Typische debiet/waterpeilvariatie voor de getijcentrale Brouwersdam”. Sent by email dated on 26-11-2013.

3 Email communation between Deltares and authorites of Province of Zeeland. Dated on 27 January 2014.

7.2 Reference literature

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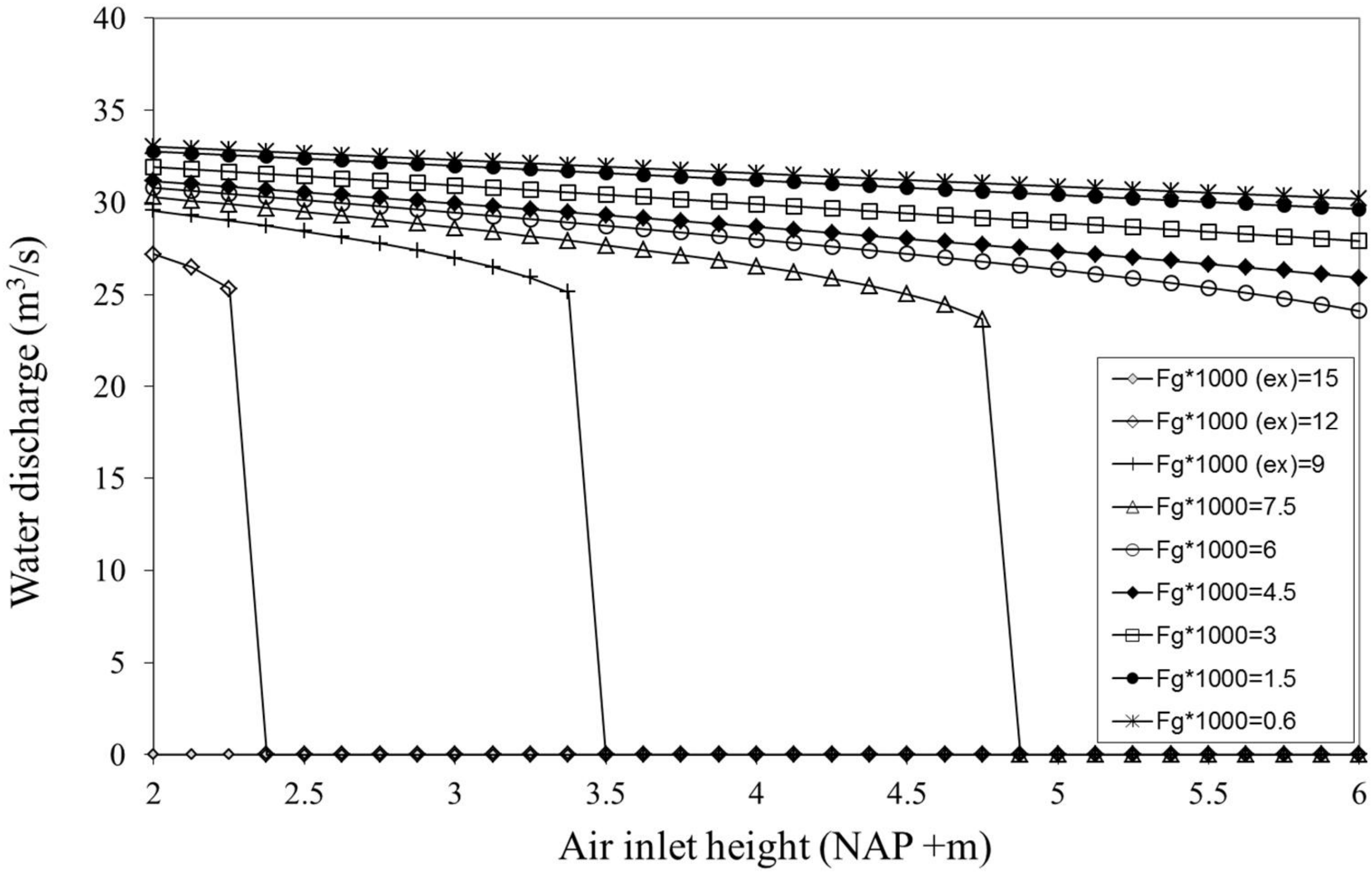
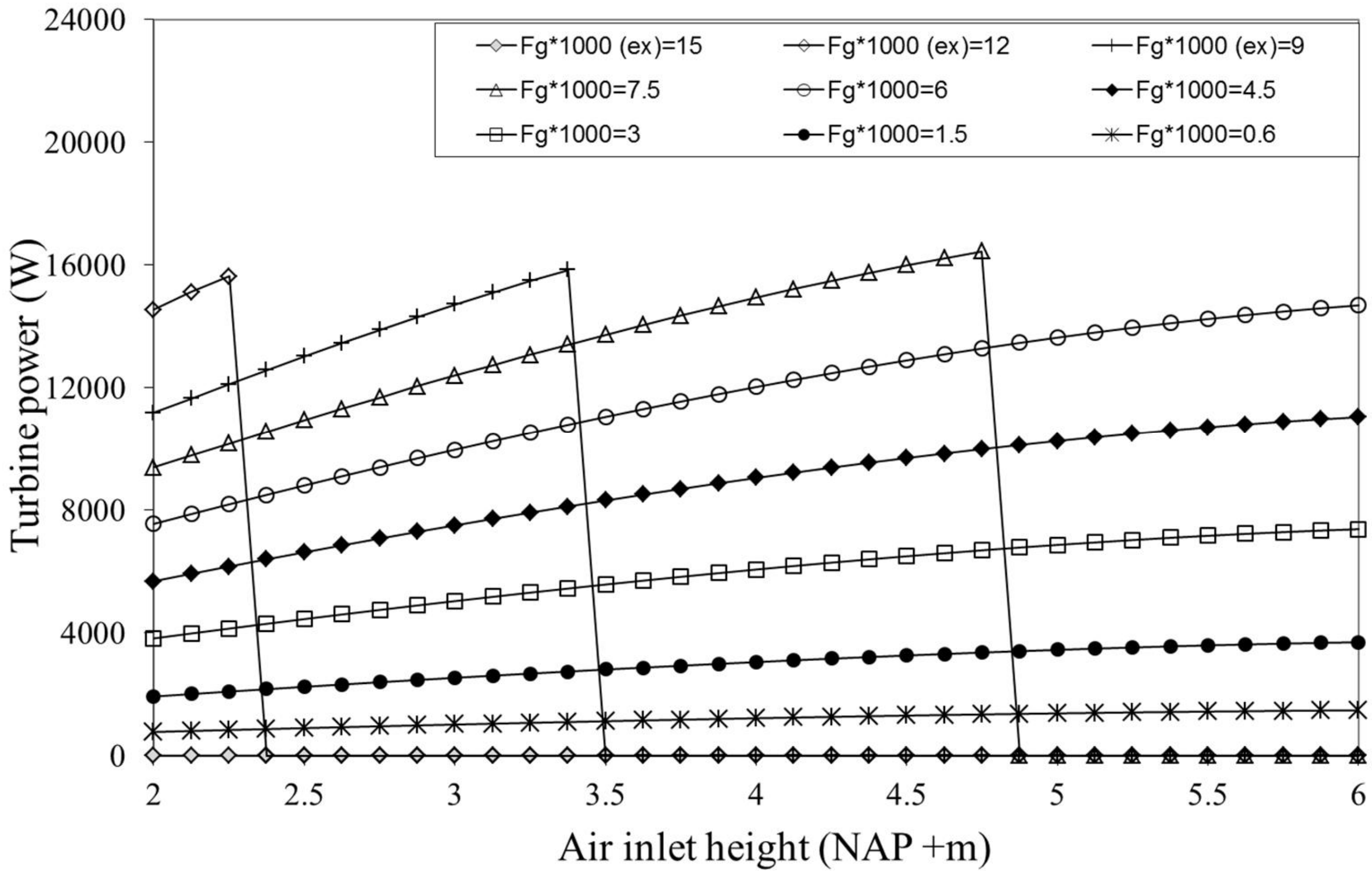
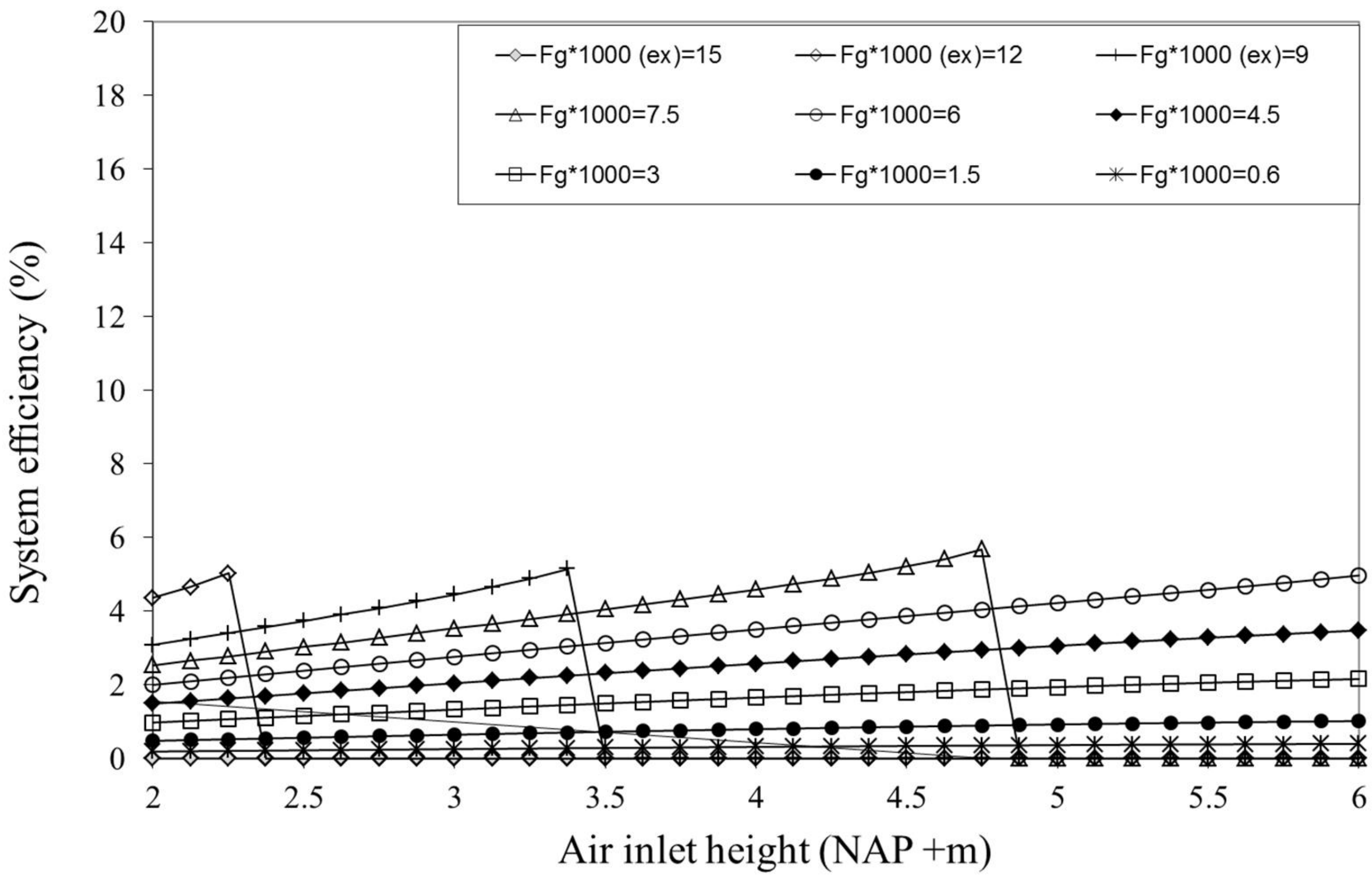
A Inverted U tube

A.1 Water level case 1 and diameter 3.2 m.



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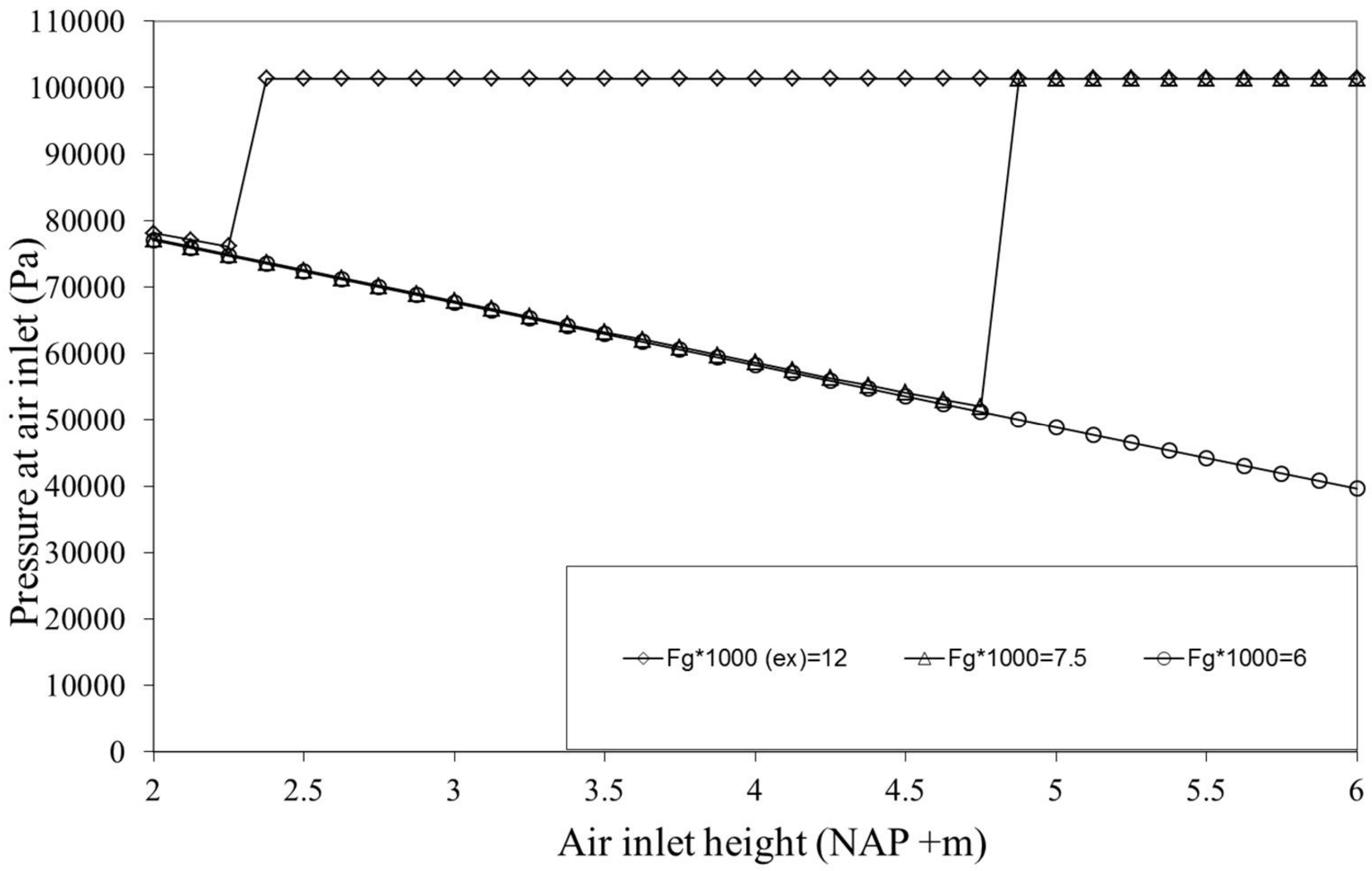
Fg*1000 (ex)=12
Fg*1000 (ex)=9
Fg*1000=7.5
Fg*1000=6
Fg*1000=4.5
Fg*1000=3
Fg*1000=1.5
Fg*1000=0.6
0
2
4
6
8
10
12
14
16
18
20
2
2.5
3
3.5
4
4.5
5
5.5
6


Pressure at air inlet (Pa)
Air inlet height (NAP +m)
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Fg*1000=7.5
Fg*1000=6
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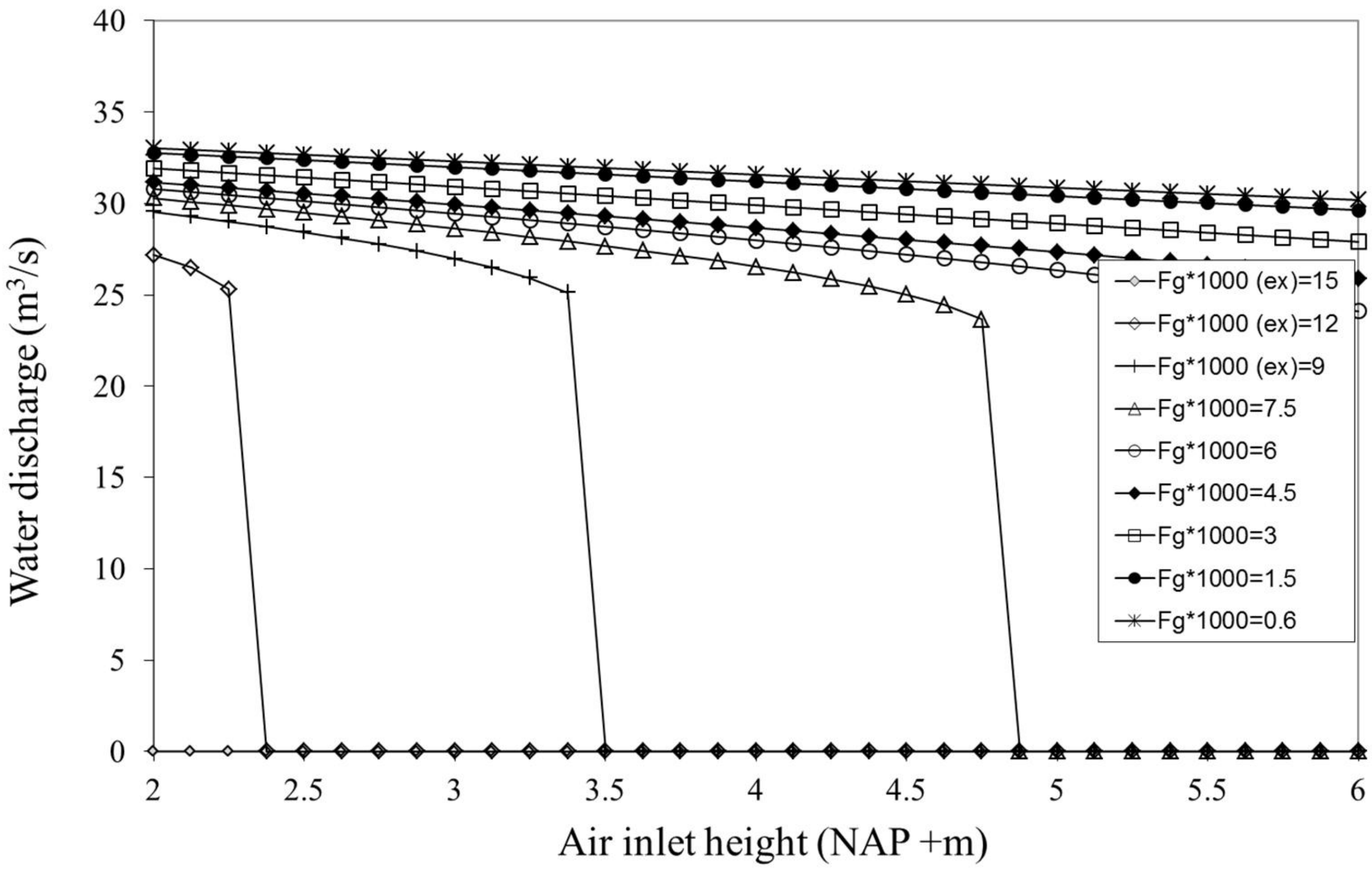
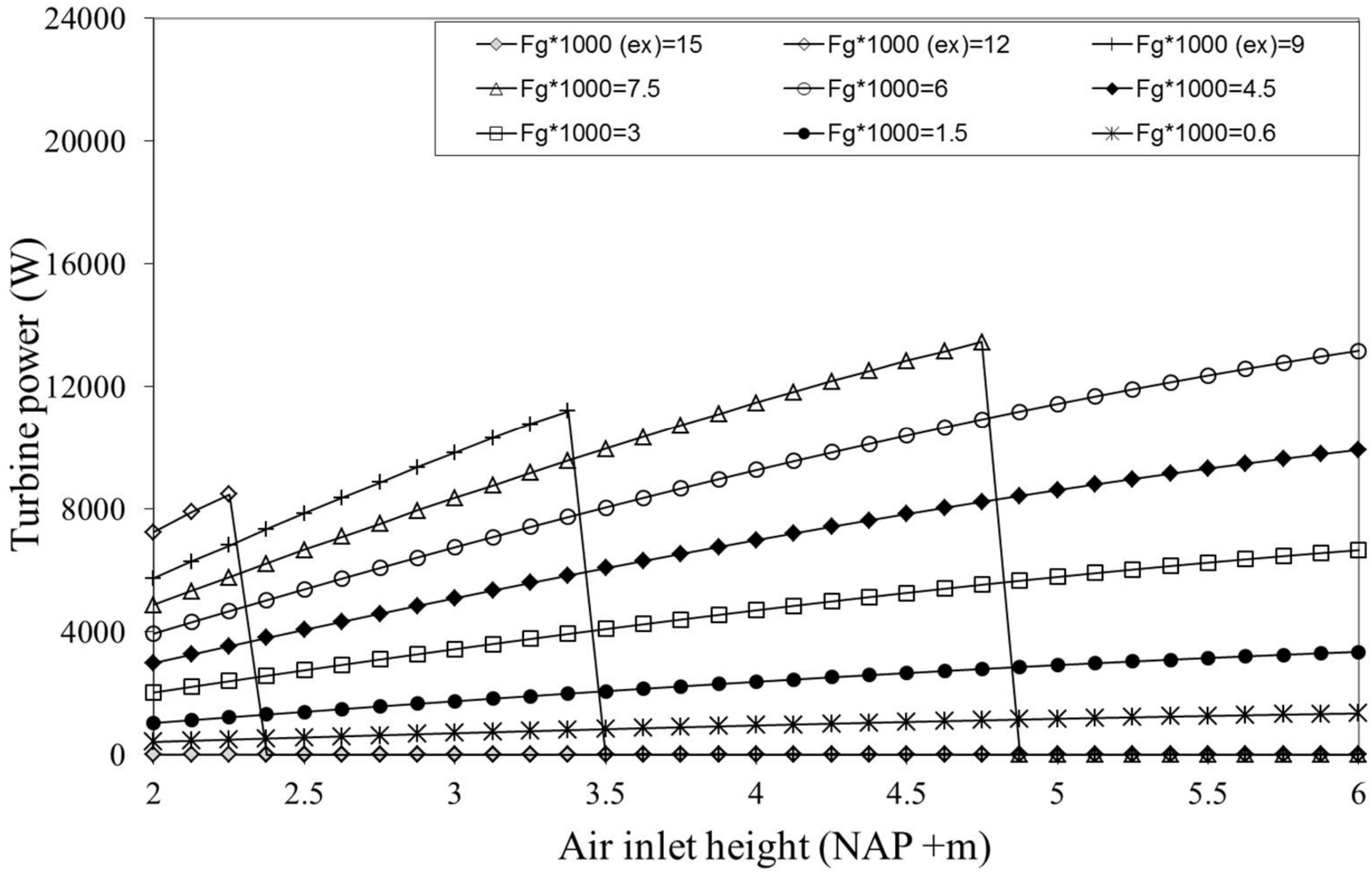
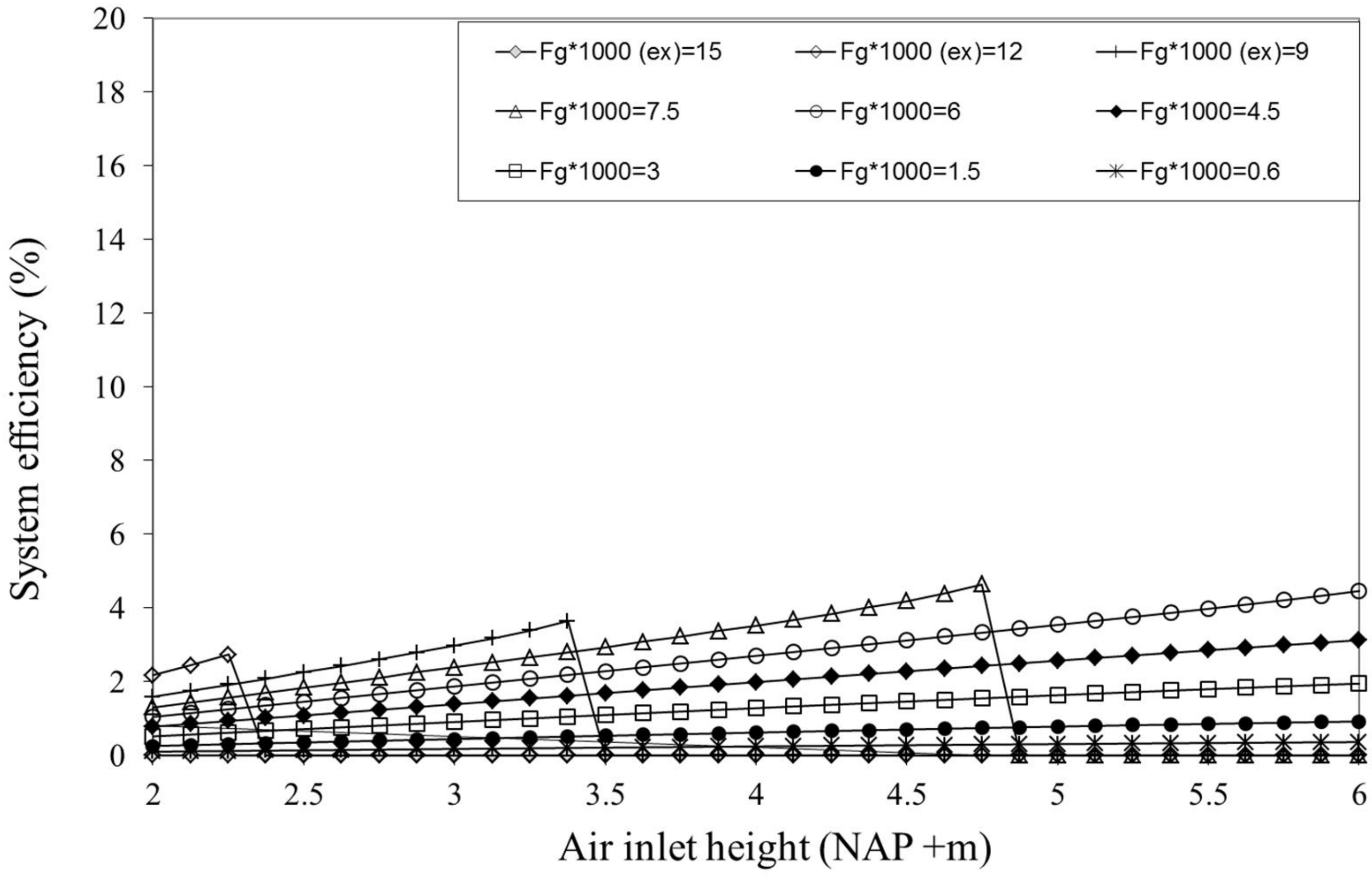
10000
20000
30000
40000
50000
60000
70000
80000
90000
100000
110000
2
2.5
3
3.5
4
4.5
5
5.5
6

A.2 Water level case 2 and diameter 3.2 m.






System efficiency (%)
Air inlet height (NAP +m)
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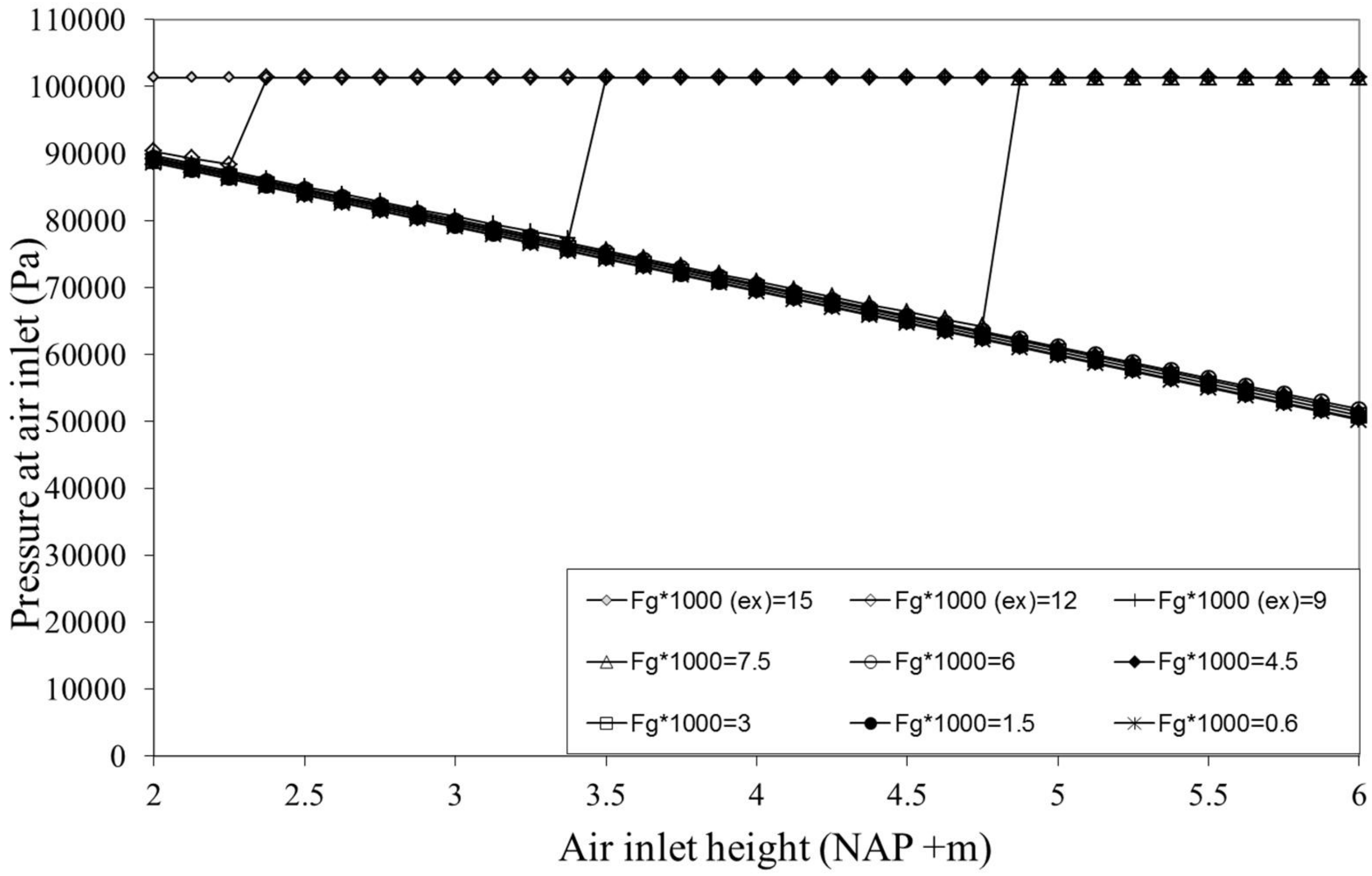
Fg*1000 (ex)=12
Fg*1000 (ex)=9
Fg*1000=7.5
Fg*1000=6
Fg*1000=4.5
Fg*1000=3
Fg*1000=1.5
Fg*1000=0.6


Pressure at air inlet (Pa)
Air inlet height (NAP +m)
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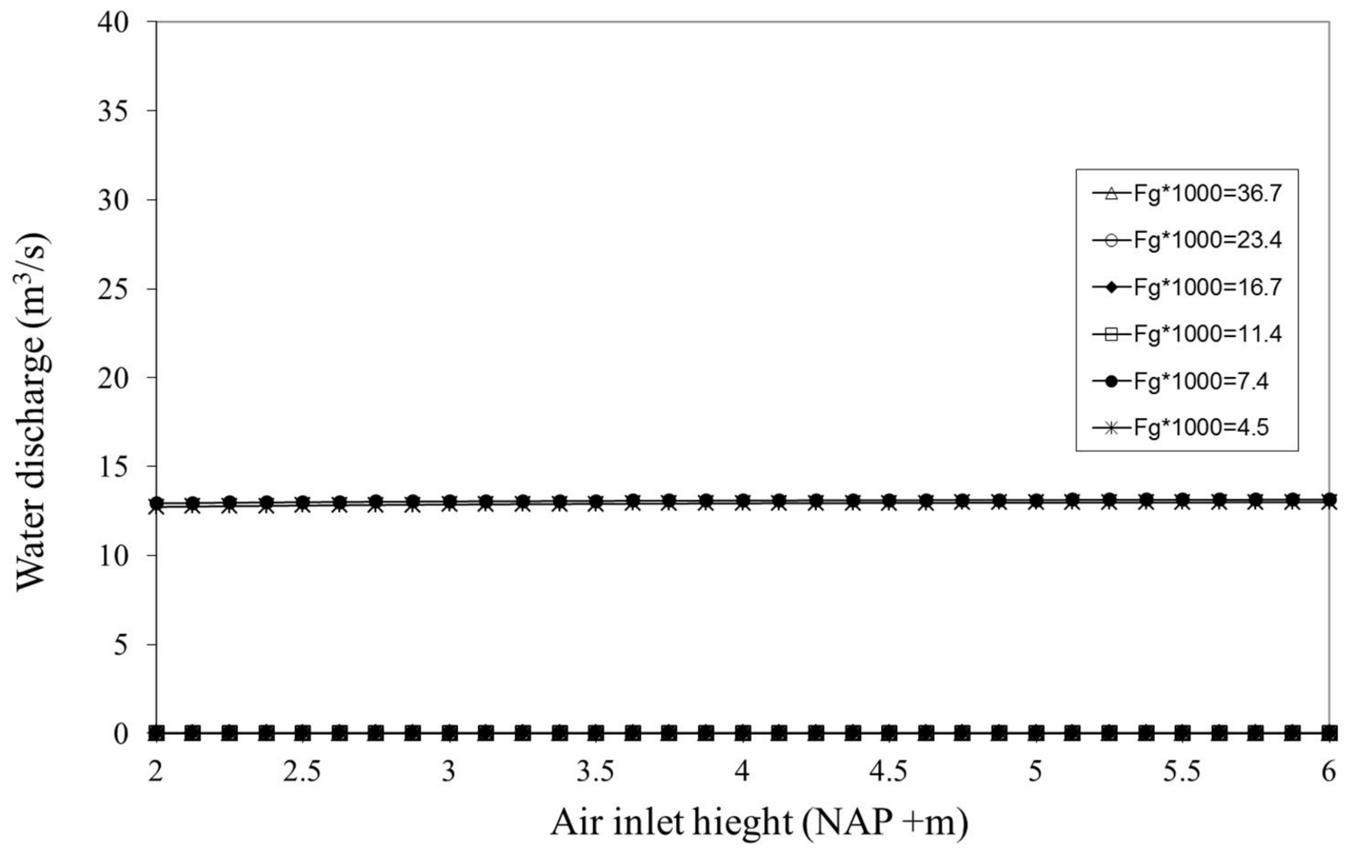
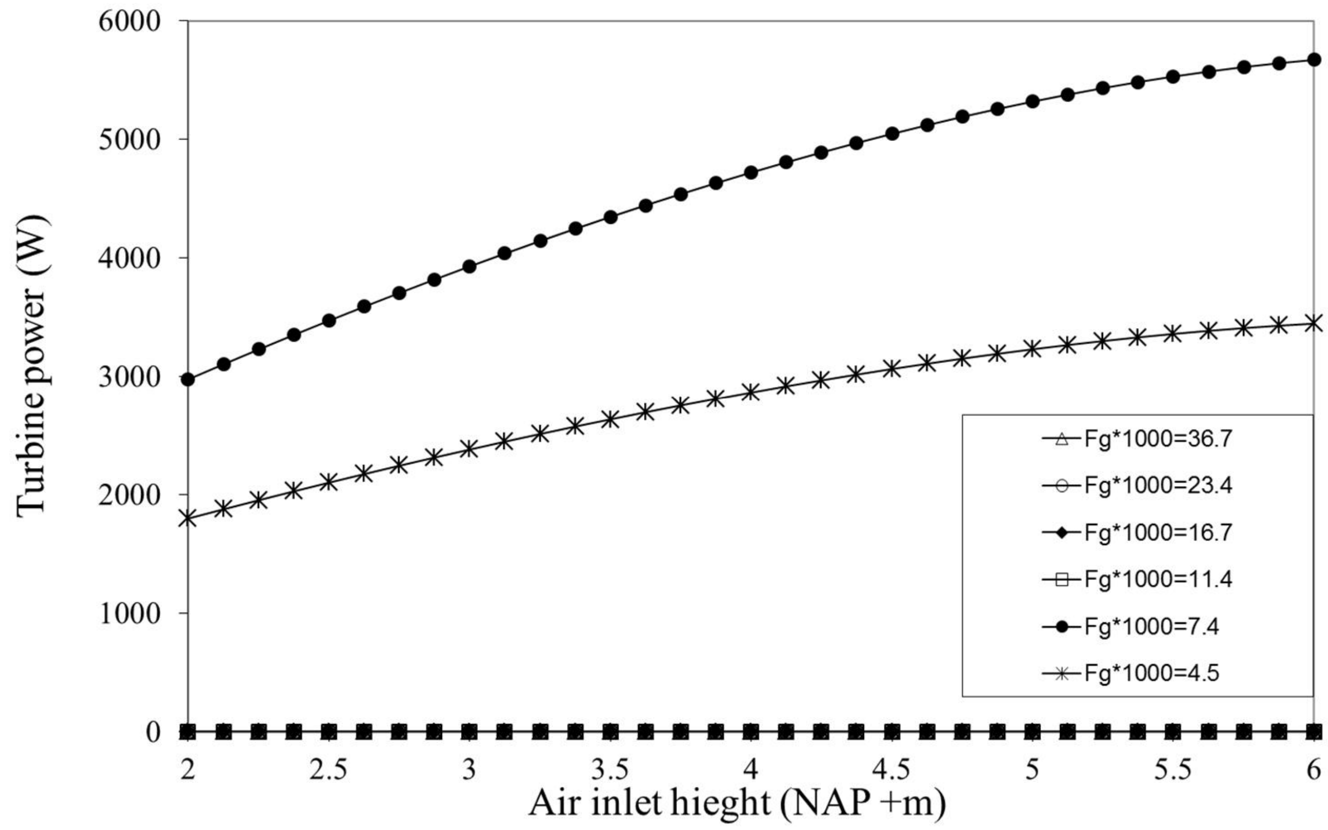
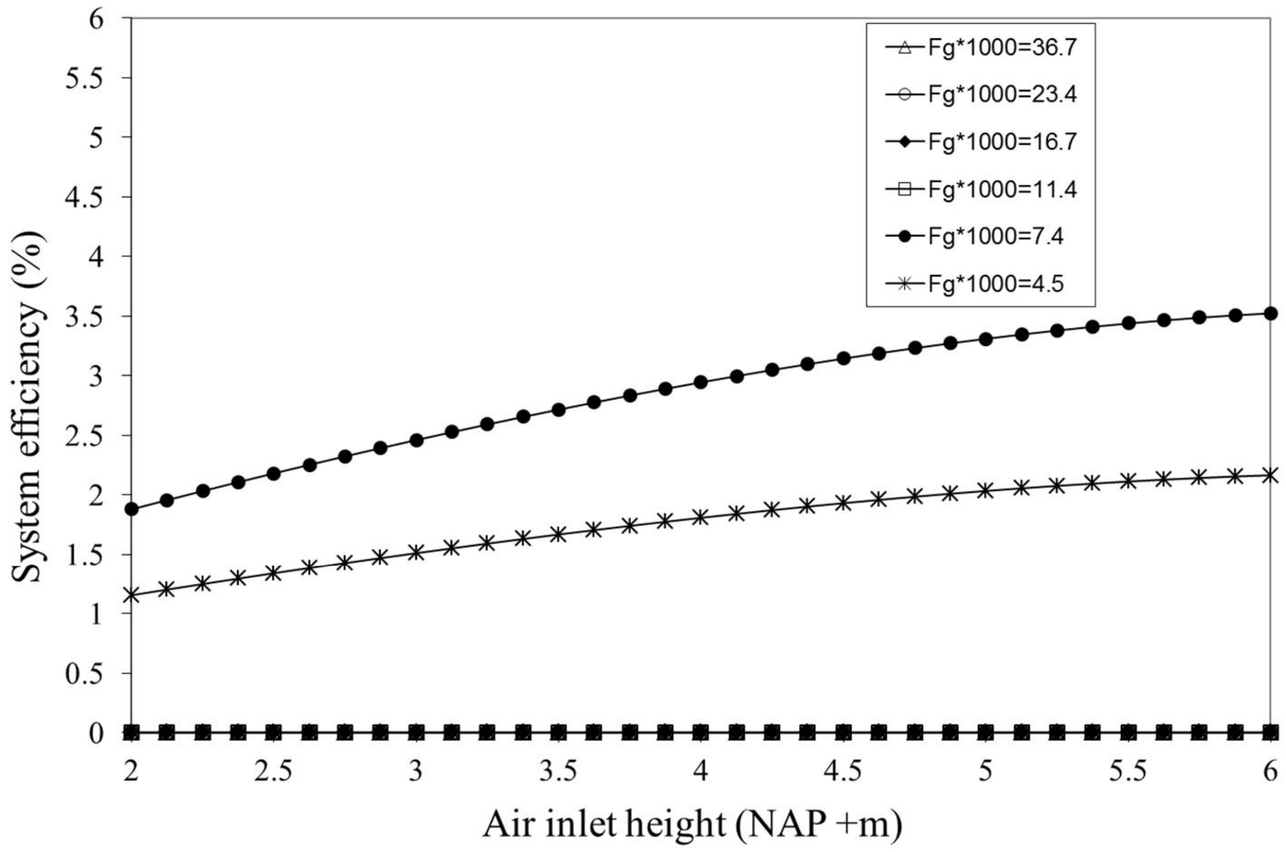
Fg*1000 (ex)=12
Fg*1000 (ex)=9
Fg*1000=7.5
Fg*1000=6
Fg*1000=4.5
Fg*1000=3
Fg*1000=1.5
Fg*1000=0.6

B Flakkeese Spuisluis

B.1 Water level case 1 and diameter 2 m.






System efficiency (%)
Air inlet height (NAP +m)
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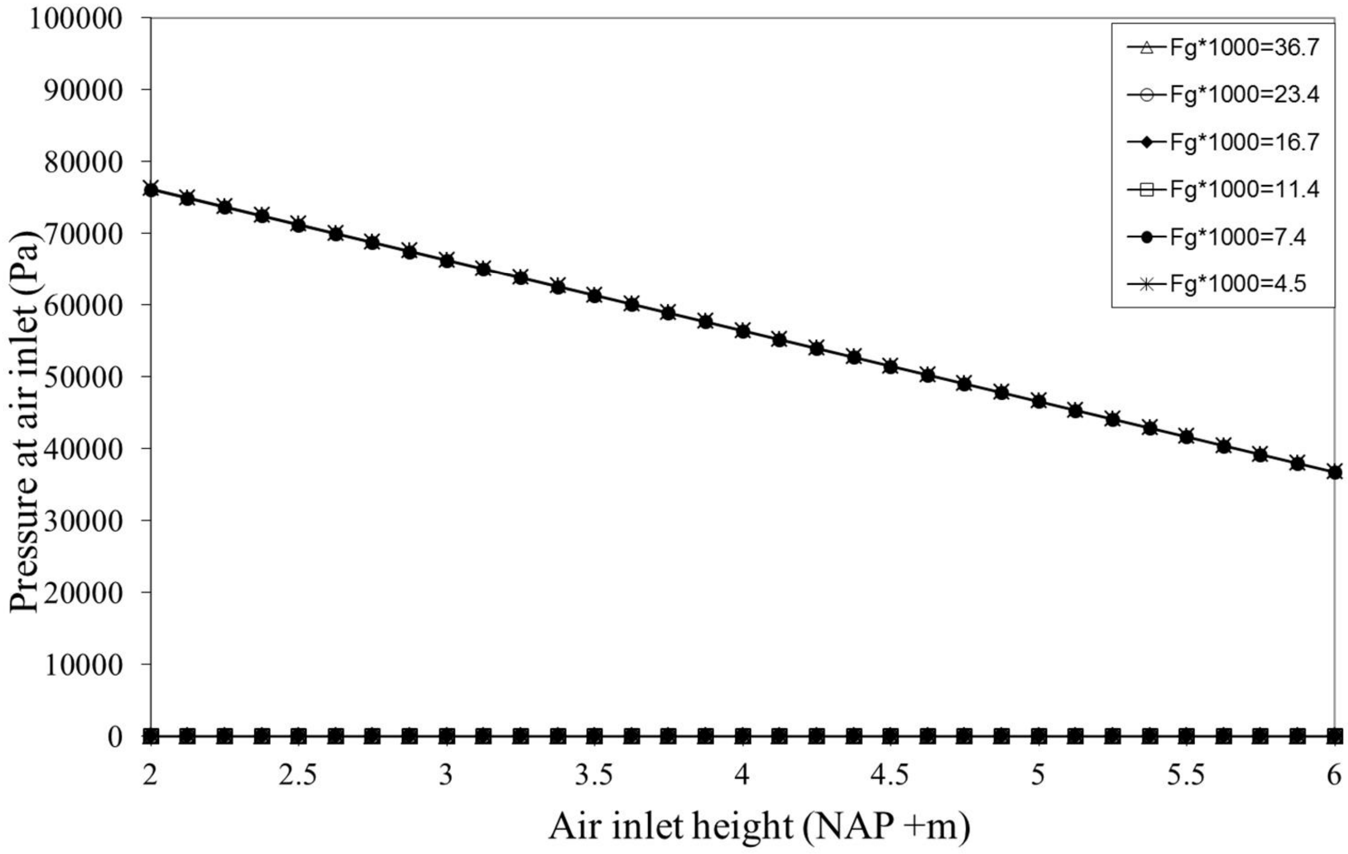
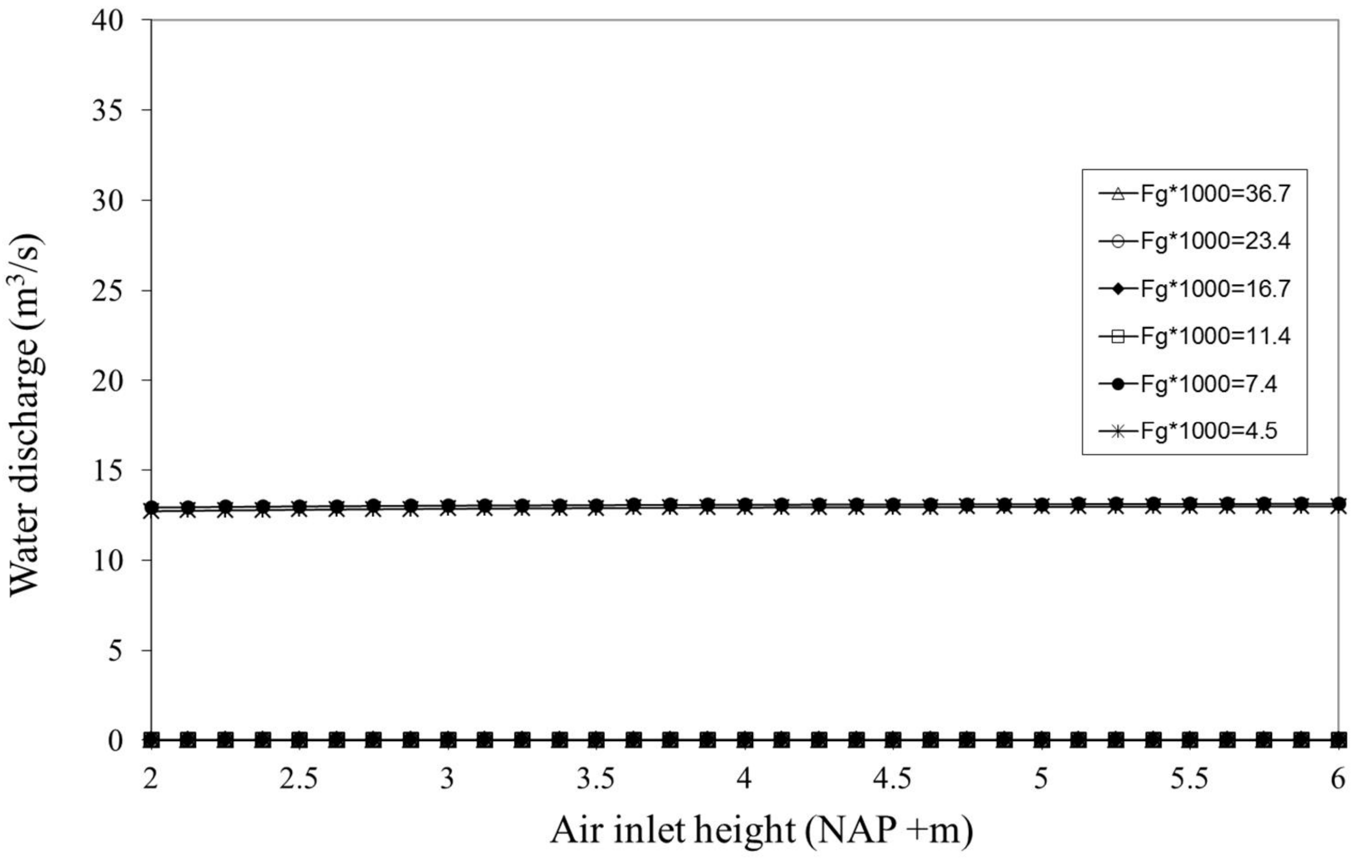
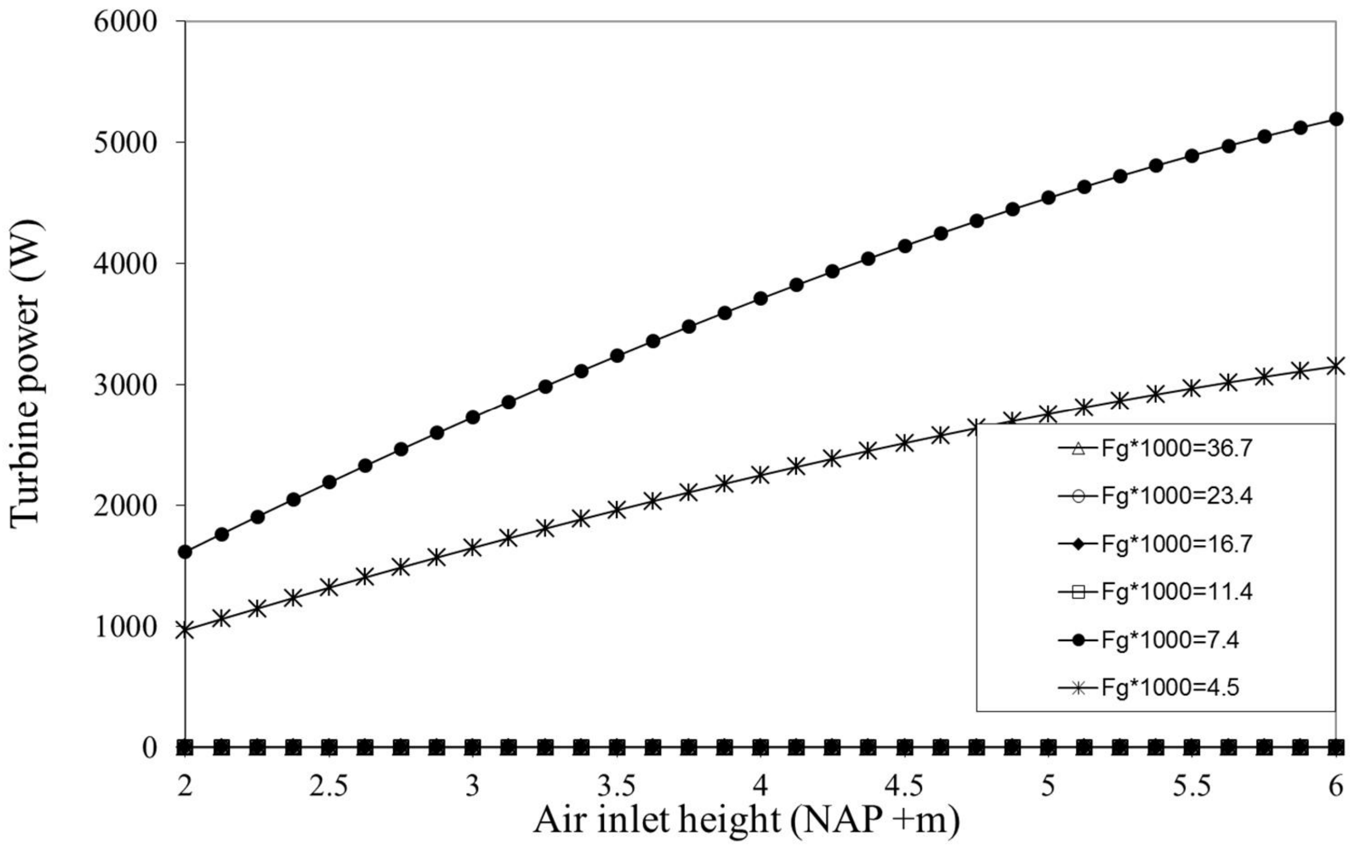
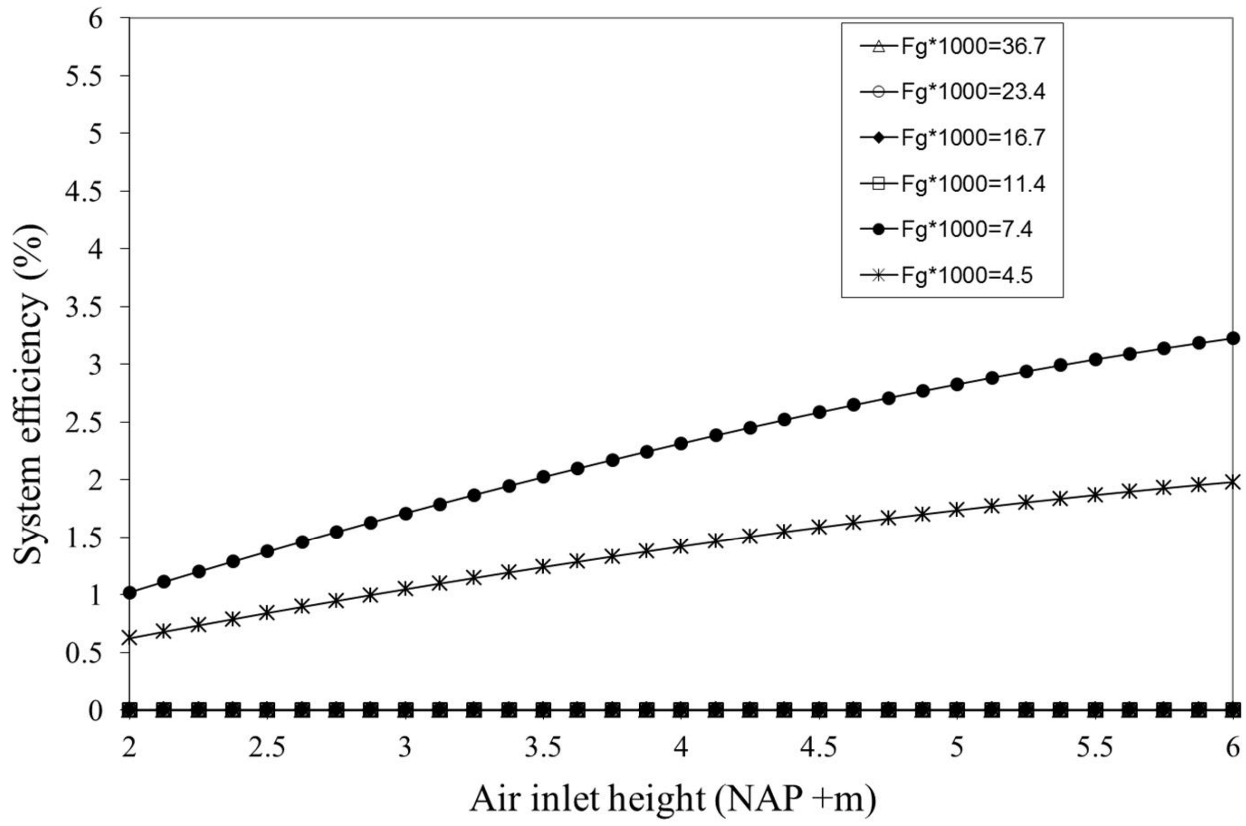
Fg*1000=23.4
Fg*1000=16.7
Fg*1000=11.4
Fg*1000=7.4
Fg*1000=4.5
0
0.5
1
1.5
2
2.5
3
3.5
4
4.5
5
5.5
6
2
2.5
3
3.5
4
4.5
5
5.5
6


Pressure at air inlet (Pa)
Air inlet height (NAP +m)
Fg*1000=36.7
Fg*1000=23.4
Fg*1000=16.7
Fg*1000=11.4
Fg*1000=7.4
Fg*1000=4.5
0
10000
20000
30000
40000
50000
60000
70000
80000
90000
100000
2
2.5
3
3.5
4
4.5
5
5.5
6

B.2 Water level case 2 and diameter 2 m.







System efficiency (%)
Air inlet height (NAP +m)
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Fg*1000=16.7
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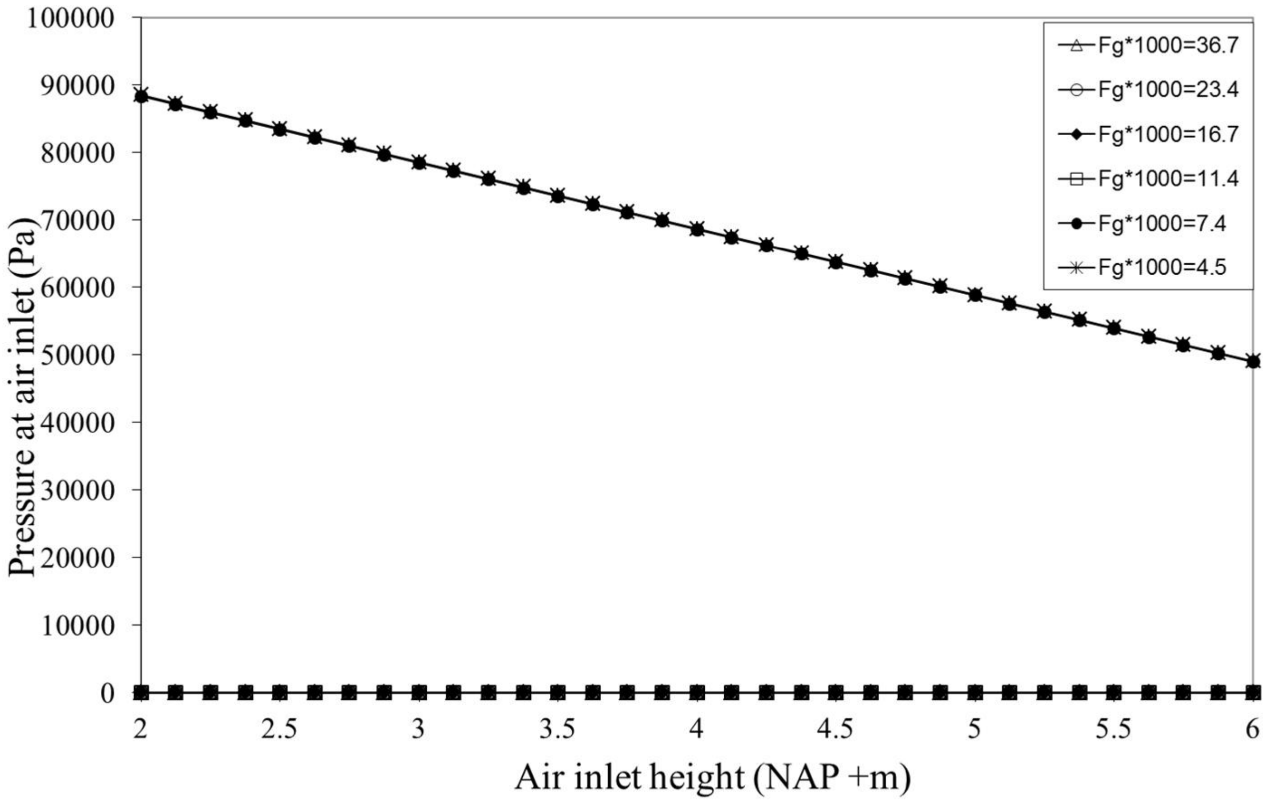
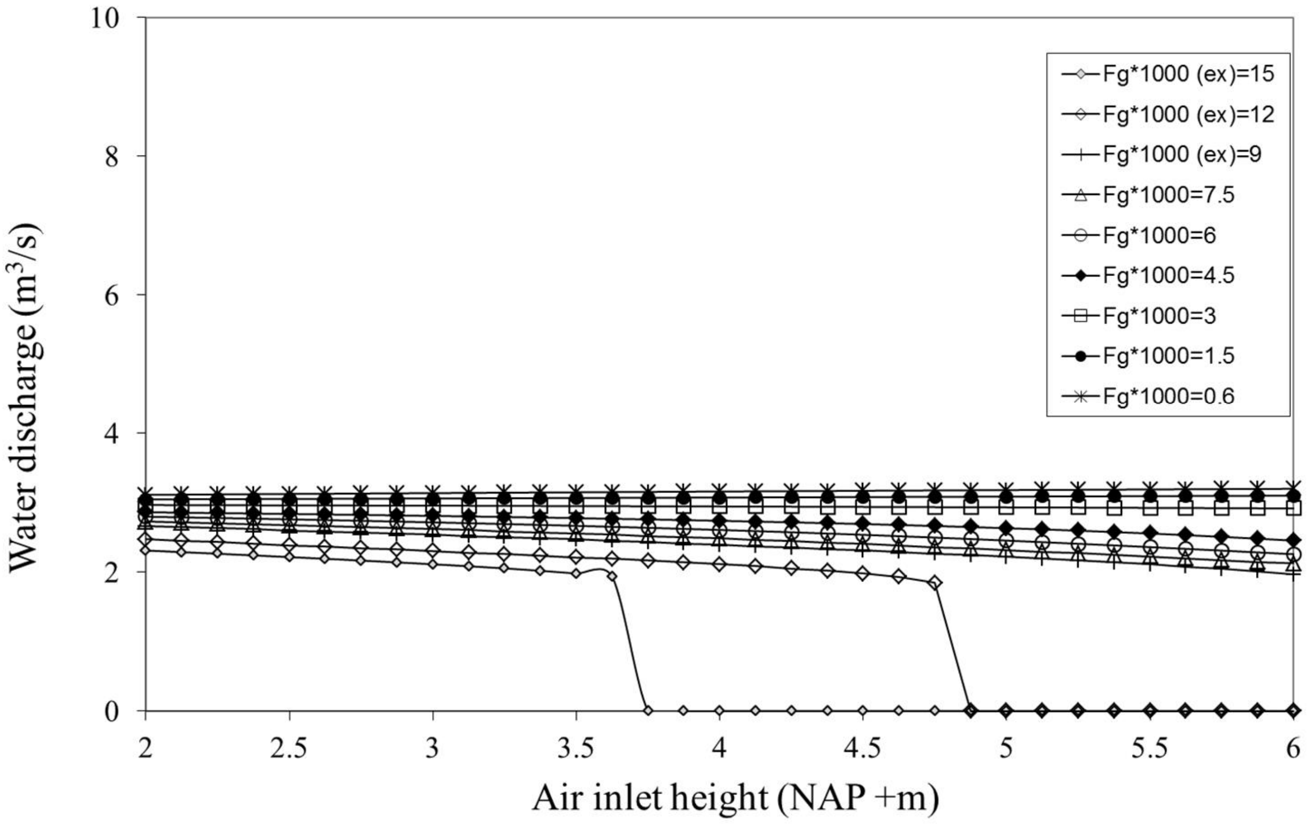
Fg*1000=4.5

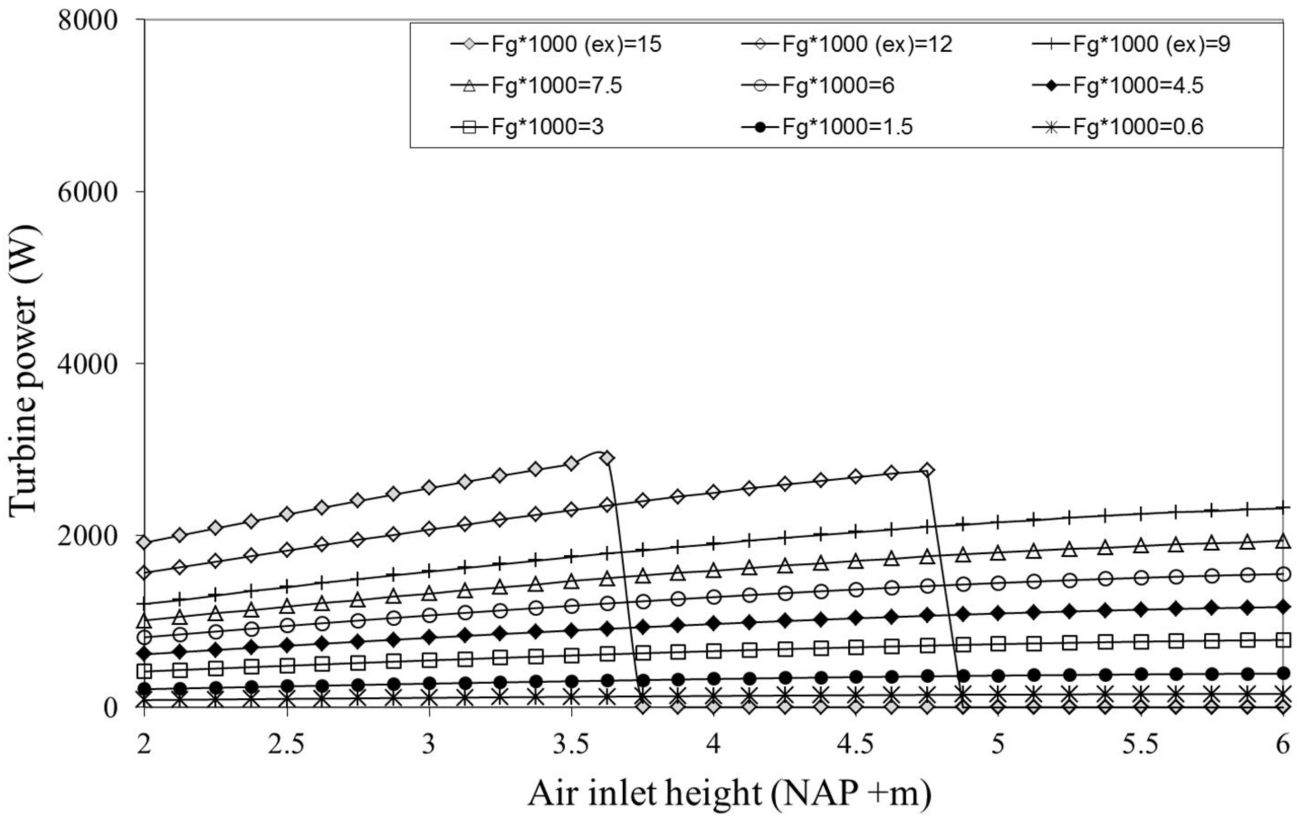


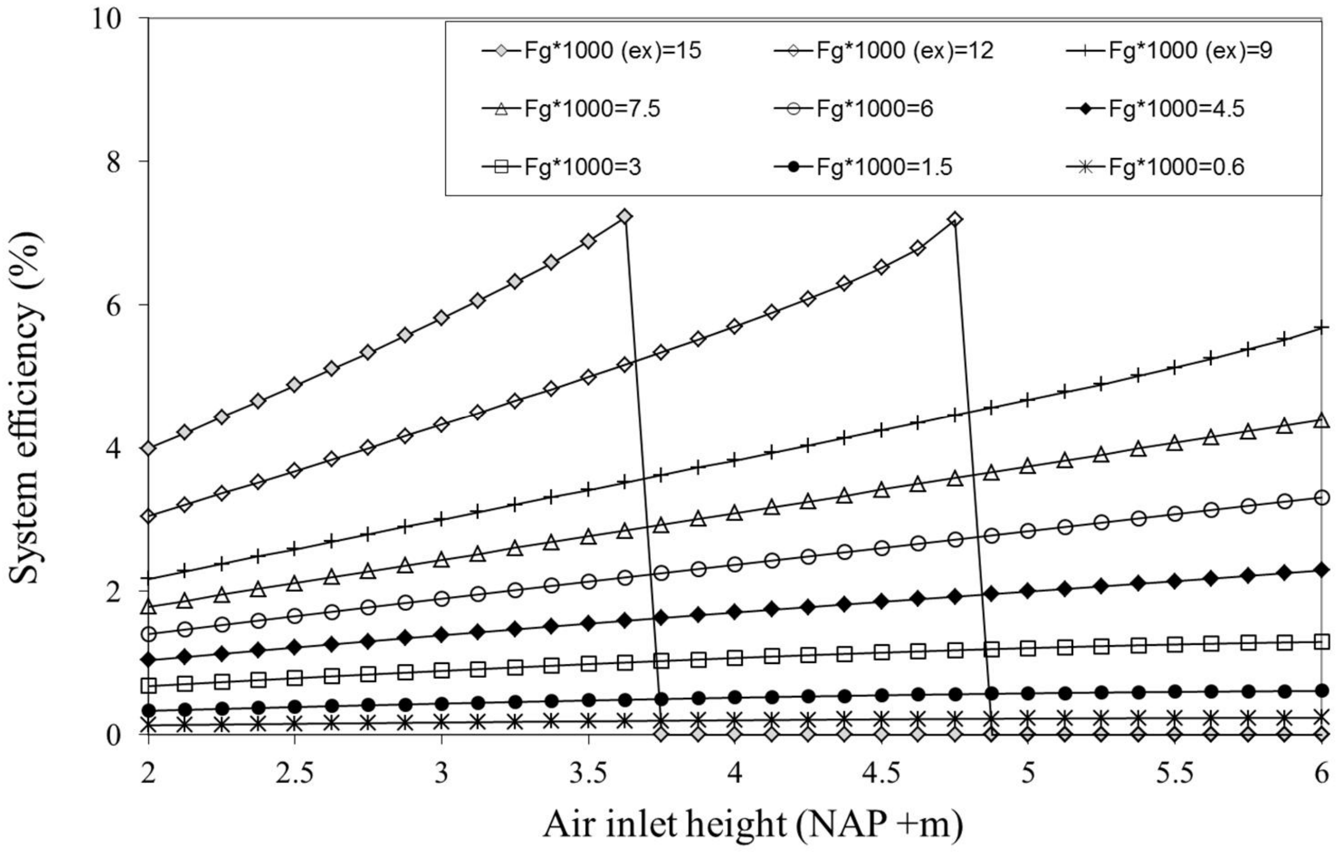
Pressure at air inlet (Pa)
Air inlet height (NAP +m)
Fg*1000=36.7
Fg*1000=23.4
Fg*1000=16.7
Fg*1000=11.4
Fg*1000=7.4
Fg*1000=4.5

C Optimum design

C.1 Inverted U tube siphon (water level case 1, diameter 1.3 m)



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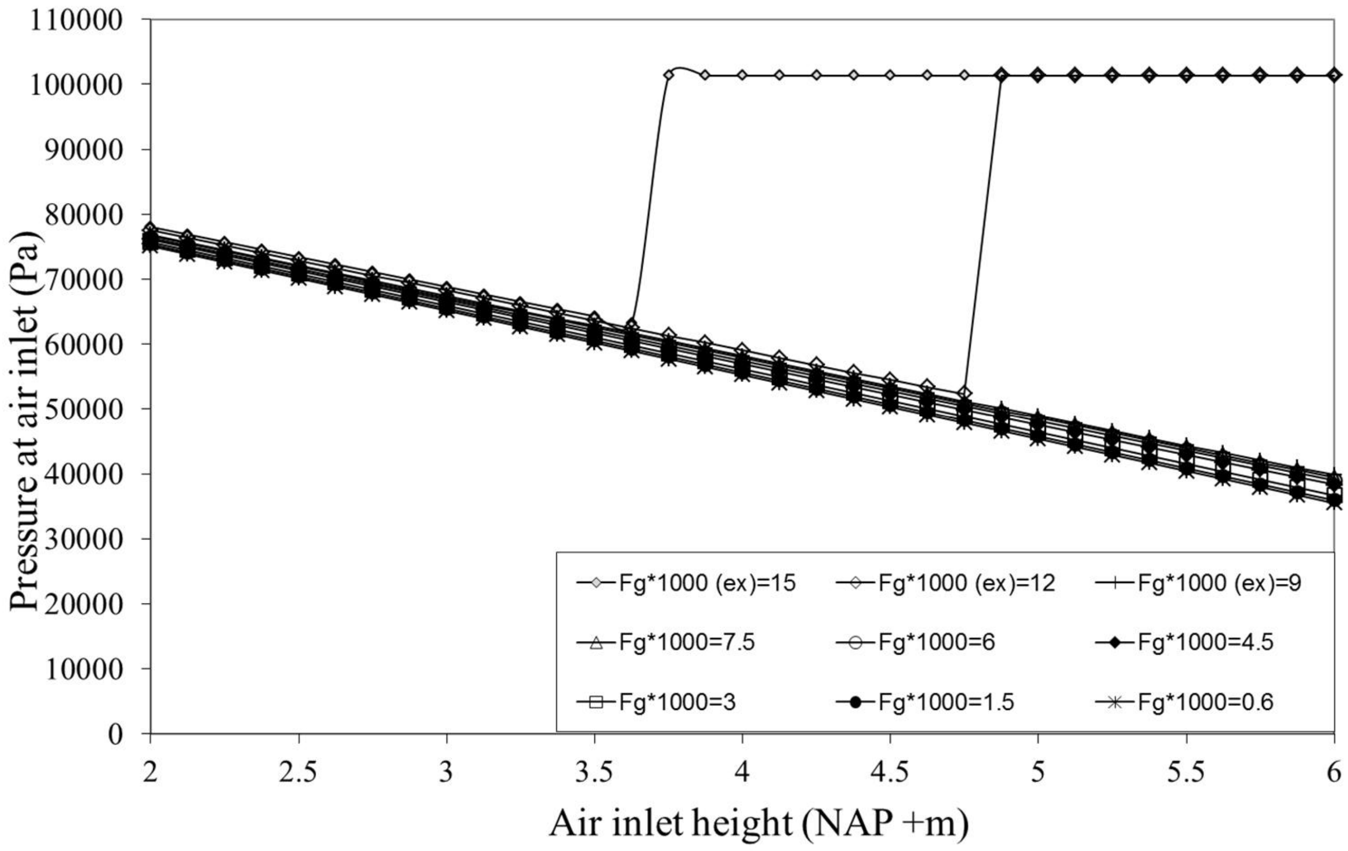
Fg*1000 (ex)=9
Fg*1000=7.5
Fg*1000=6
Fg*1000=4.5
Fg*1000=3
Fg*1000=1.5
Fg*1000=0.6
System efficiency (%)
Air inlet height (NAP +m)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Fg*1000 (ex)=9
Fg*1000=7.5
Fg*1000=6
Fg*1000=4.5
Fg*1000=3
Fg*1000=1.5
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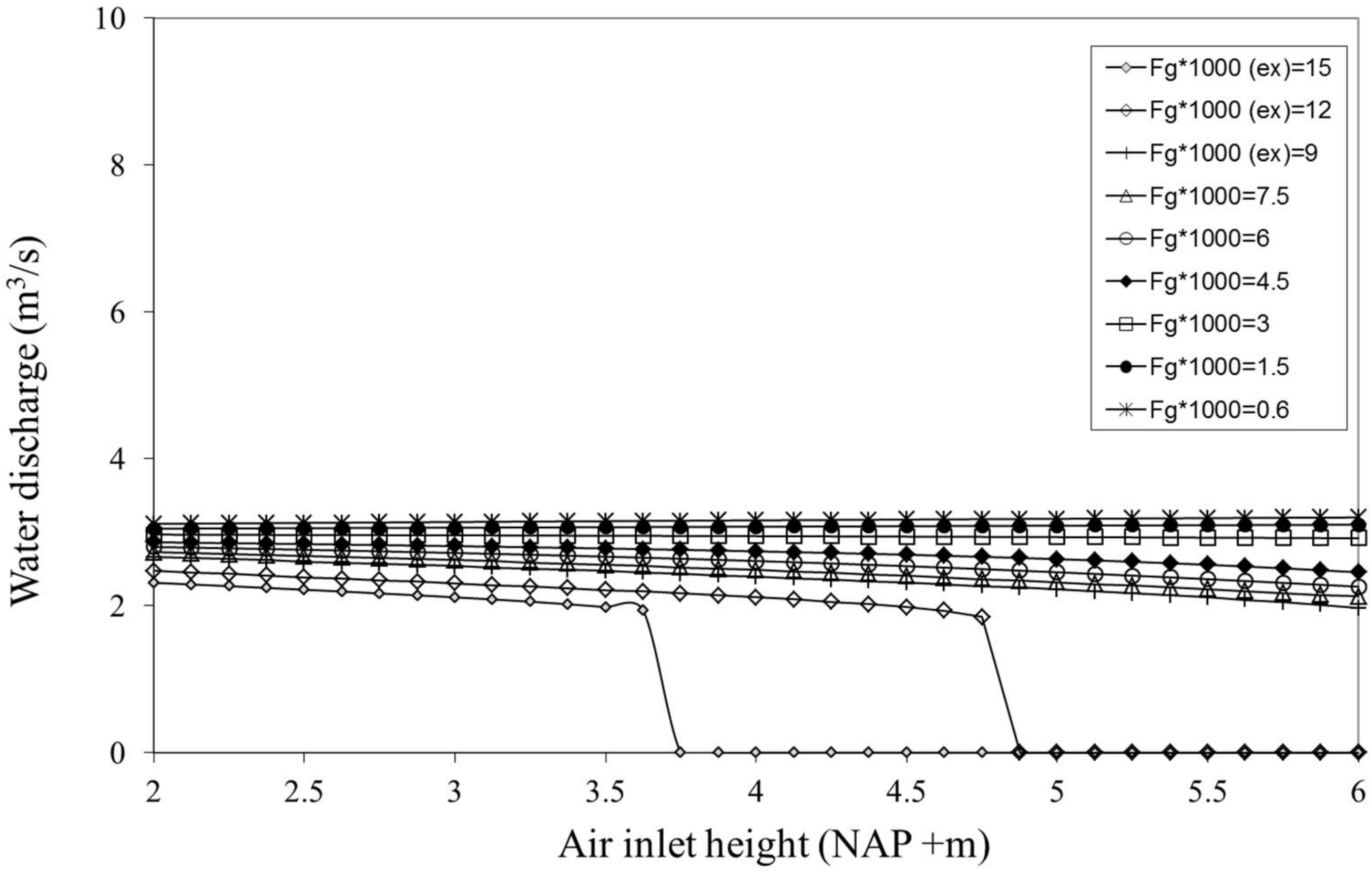
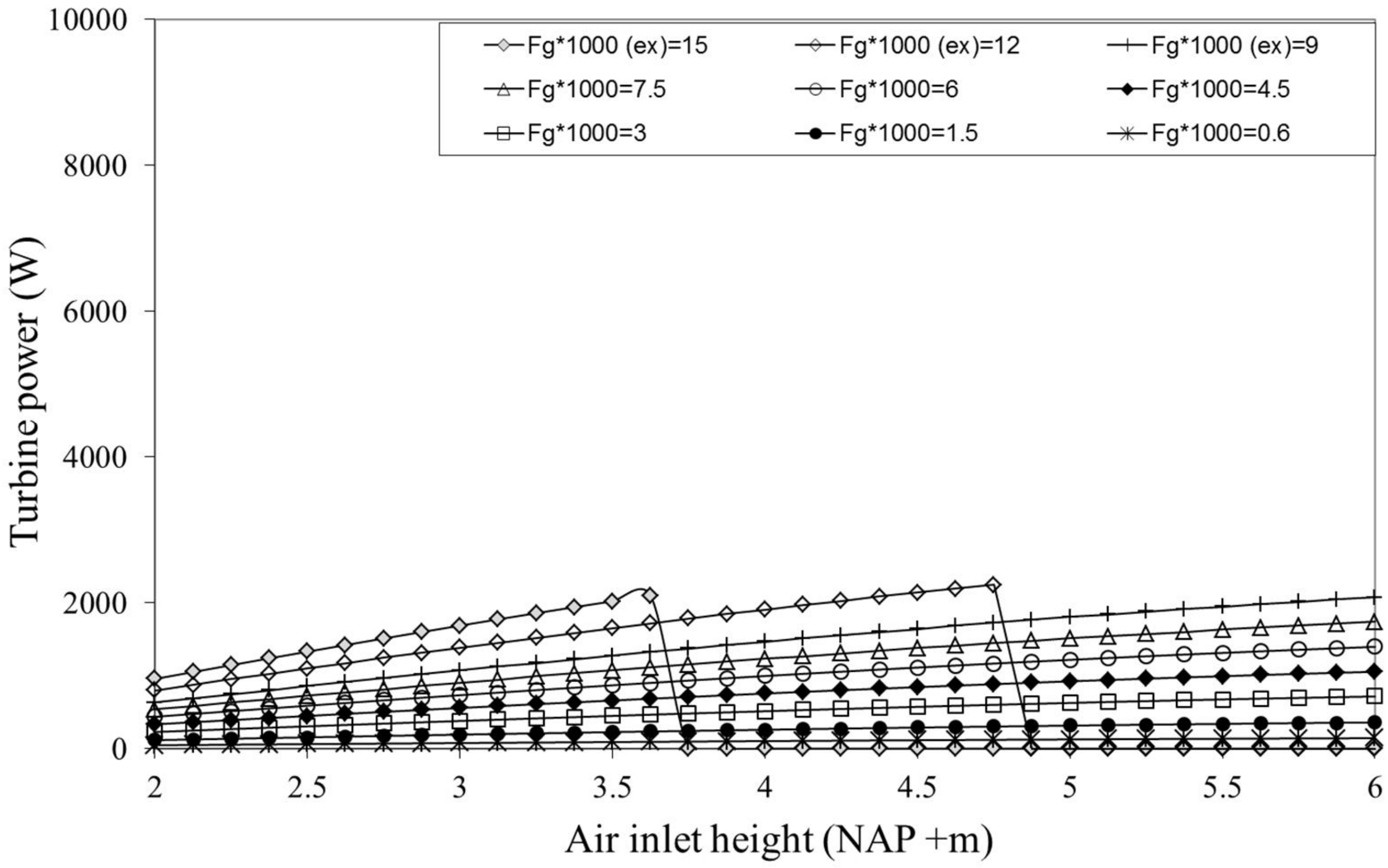
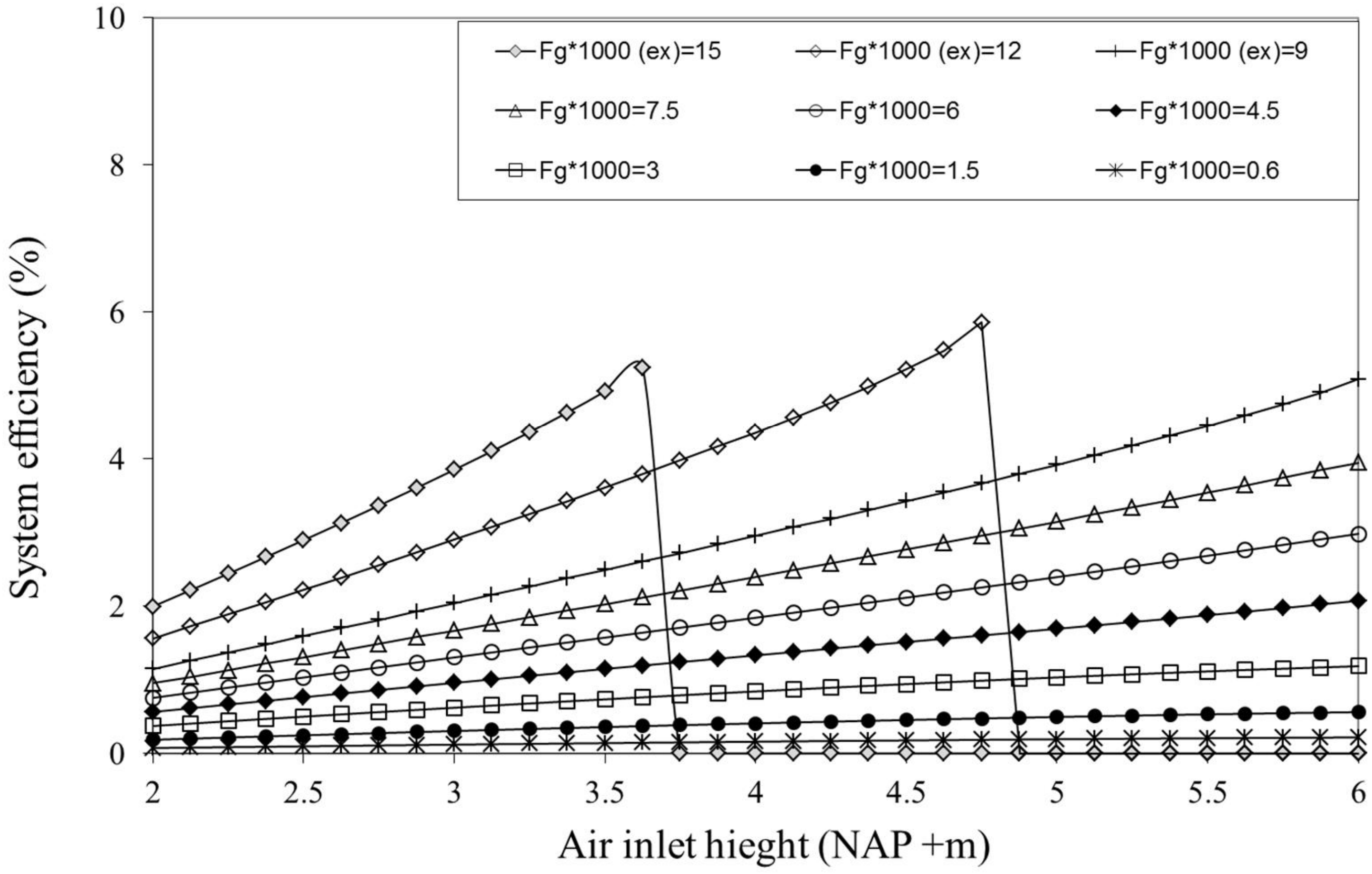
Pressure at air inlet (Pa)
Air inlet height (NAP +m)

C.2 Inverted U tube siphon (water level case 2, diameter 1.3 m)



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Fg*1000 (ex)=12
Fg*1000 (ex)=9
Fg*1000=7.5
Fg*1000=6
Fg*1000=4.5
Fg*1000=3
Fg*1000=1.5
Fg*1000=0.6
System efficiency (%)
Air inlet hieght (NAP +m)
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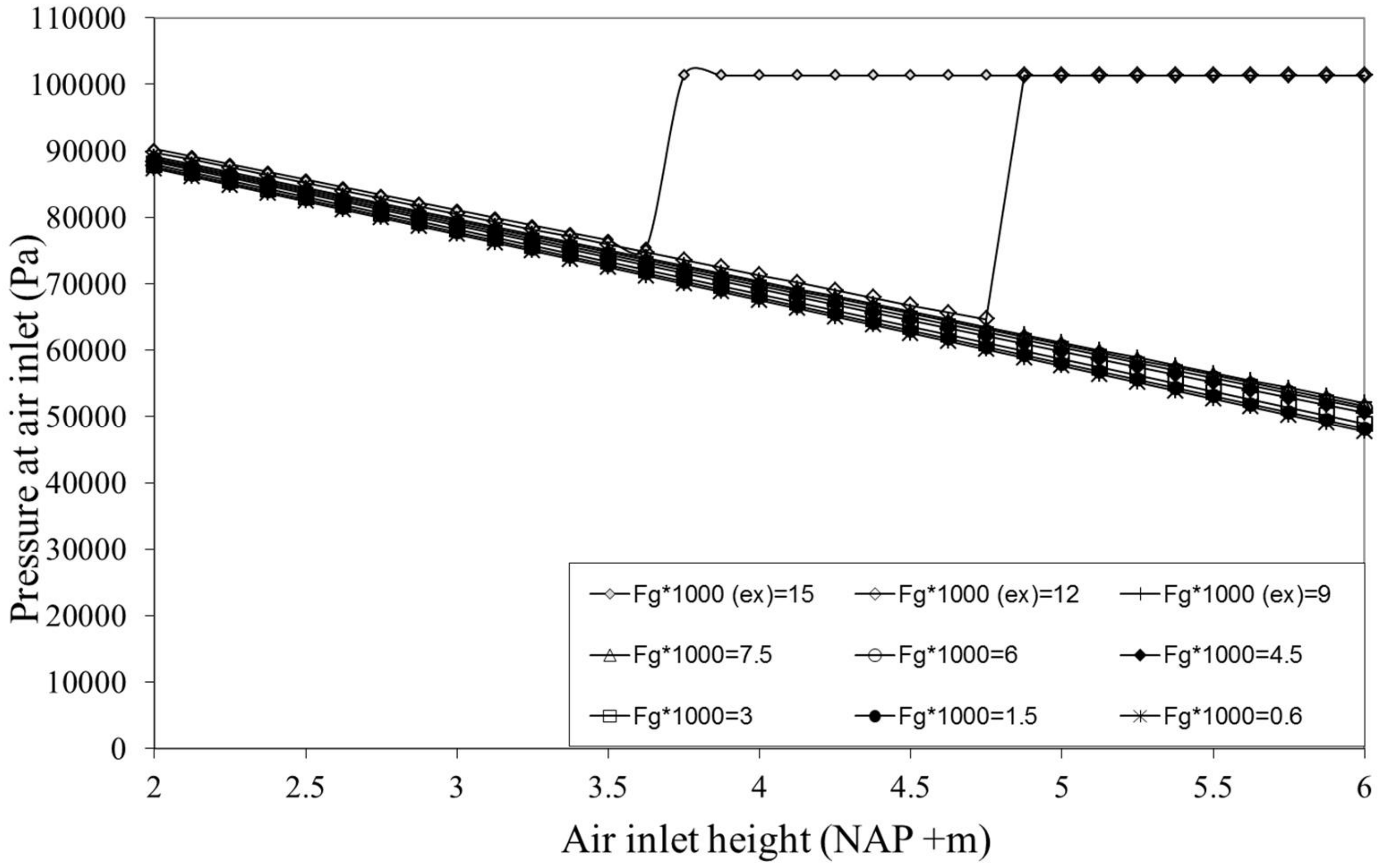
2
4
6
8
10
2
2.5
3
3.5
4
4.5
5
5.5
6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Fg*1000 (ex)=12
Fg*1000 (ex)=9
Fg*1000=7.5
Fg*1000=6
Fg*1000=4.5
Fg*1000=3
Fg*1000=1.5
Fg*1000=0.6
Pressure at air inlet (Pa)
Air inlet height (NAP +m)
0
10000
20000
30000
40000
50000
60000
70000
80000
90000
100000
110000
2
2.5
3
3.5
4
4.5
5
5.5
6