Turbine development for the Wave Dragon wave energy converter

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Abstract

The Wave Dragon is a wave energy converter of the overtopping type which has been developed by a European consortium during the last five years. It uses a set of low head water turbines as power takeoff system. In the course of a two year research programme the basic conception of the turbines has been established, the turbine geometry has been designed and a model turbine has been built and tested. Furthermore, a software simulation has been conceived that allows the operation of the Wave Dragon in different sea conditions to be simulated, thus permitting a tailoring of the turbines as well as a forecast of the annual energy production for installation in a proposed location.

The paper at hand is giving an introduction into wave energy converters in general and is describing the methods applied in as well as the results obtained from the turbine development program.

1. Introduction

Wave energy is a source of renewable energy holding a great potential, but it has so far not been commercially exploited. A wide variety of functional principles and designs for wave energy converters has been proposed, which may be divided into three main categories:

- point absorber devices
- oscillating water column devices and
- overtopping devices.

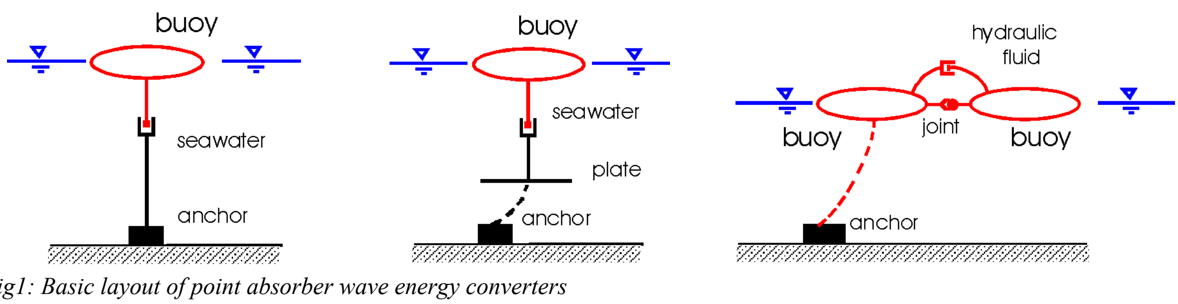


Fig1: Basic layout of point absorber wave energy converters

The point absorber devices consist of either a single or an arrangement of multiple buoyant bodies floating on the surface of the waves, the movement of which is used to drive a generator through a hydraulic high pressure system, see Fig. 1. A major problem with these is the limited life of the joints and hinges as well as the hydraulic cylinders, which are subjected to about 5 millions of load cycles a year.

The devices in the second category, the so-called OWC devices, work on the principle shown in Fig. 1.2: The pressure fluctuations from the passing waves lead to oscillations of the water column enclosed in the main body of the device. The air forced in and out of the body is driving an air turbine and a generator. The main difficulty with these devices is the reciprocating direction of the turbine flow. The current solution is to use a Wells turbine, which turns in the same direction regardless of the flow direction but has a relatively low efficiency.

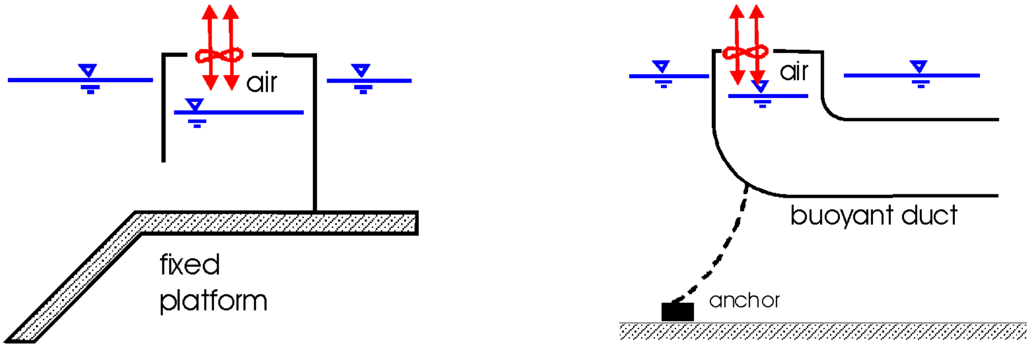


Fig. 2: Basic layout of Oscillating Water Column wave energy converters

The devices in the third category are based on overtopping. In these devices, the kinetic and potential energy of the waves is utilized to drive the water up a ramp, resulting in an overtopping flow into a reservoir situated above the sea, see Fig. 2. The potential energy of the water in the reservoir is exploited by releasing it back into the sea through a water turbine.

So far none of the many wave energy converters that have already been proposed has proven to be feasible in construction and efficient long-term operation. Experience has shown that most of the designs failed after a relatively short period, mostly due to mechanical problems in the structure itself or in the power takeoff systems. There are, however, a number of promising devices that are now being tested in small prototype scale.

One of these is the Wave Dragon, a slack moored, floating converter of the overtopping type. It has been developed and tested in 1:50 model scale during the last 5 years by an international consortium sponsored by the Danish Wave Energy Programme and the European Community. In autumn 2002, a second project phase has been started, during which a 237 ton steel prototype in scale 1:4.5 has been built which is now being real sea tested for a period of three years in a Danish fjord.

2. Design and functioning of the Wave Dragon

2.1. Basic design of the hull

The Wave Dragon consists of two reflector arms and a central hull comprising a storage reservoir, see Fig. 3. The reflector arms are focusing the waves onto the doubly curved ramp on the front of the hull, where the waves run up and fill the reservoir which is situated above the mean sea level. The water in the reservoir is released back into the sea through a set of specially designed low head water turbines. Thus, in contrast to the majority of the other wave energy converters proposed hitherto, the Wave Dragon utilizes well proven and reliable hydro power technology for the power takeoff.

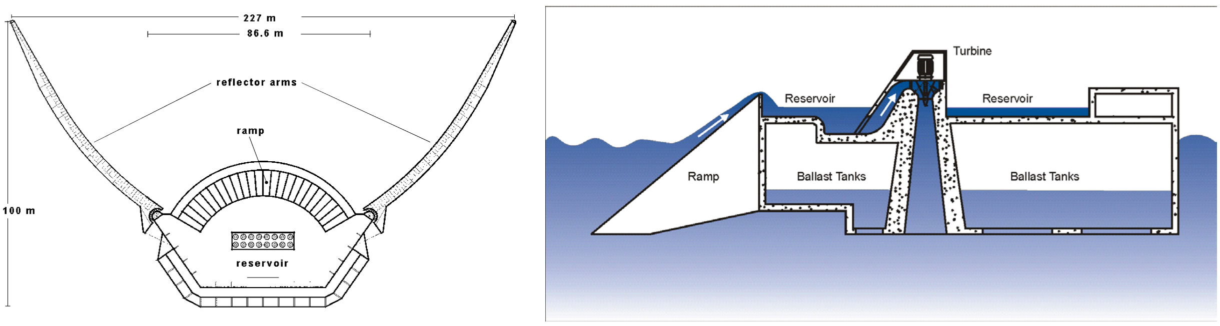


Fig. 3: Basic layout of the Wave Dragon

The shape and angle of the reflectors as well as the curvature of the ramp have been optimised for maximum capturing efficiency in wave tank tests using a 1:50 scale model, see [1, 2]. In order to achieve an efficient operation in low as well as in high waves, the crest height of the device can be adjusted by regulating the air pressure in the ballast tanks which have apertures in the bottom of the hull.

2.2. Operating conditions of the turbines

After the optimum configuration of the hull and arms had been determined, a big number of wave tank tests have been performed in order to establish a correlation giving the overtopping flow rate as a function of the sea state and the crest height, see [3]. It has been found that the overtopping can be simulated using a random process based on the following formula for the average overtopping flow rate:

$$q = 0.025 c_d \cdot \exp\left(-40 \frac{Rc}{H_s} \sqrt{\frac{S_{op}}{2\pi}}\right) \frac{L\sqrt{gH_s^3}}{\sqrt{\frac{S_{op}}{2\pi}}}$$

wherein:

c_d reduction coefficient for directional spreading [-]
Rc crest height [m]
S_{op} wave steepness, $S_{op} = \frac{H_s \cdot 2\pi}{gT_p^2}$ [-]
T_p wave peak period
L crest length [m]
H_s significant wave height [m]

As the Wave Dragon is a floating device, the calculation of the energy production in a certain sea state is not a straightforward task. The overtopping inflow depends on the wave height and the crest height, which again is influenced by the water level in the reservoir due to the immersion of the hull. The reservoir level is depending on the overtopping and on the turbine discharge, which is a function of the turbine head and speed, with the turbine head again depending on the crest height and basin level. Thus, there are multiple interdependencies between most of the parameters involved.

This makes maximising the overall efficiency or the annual energy output a very complex task, especially as a great number of parameters needs to be optimised and the effect of any parameter change has to be studied for a number of different sea states.

Fig. 4 is giving a schematic view of the whole device. Starting from a chosen value for the initial crest height Rc0, the turbine characteristics Q=f(H) and η=f(H,Q), a given regulating strategy and a certain wave height H_S, the following instantaneous quantities need to be determined:

- Crest height Rc = f(Rc0, Immersion)
- Overtopping flow rate $Q_{ov} = f(H_s, Rc)$
- Basin level $l = f(Q_{ov}, Q_{Tu}, Q_{Spil})$
- Immersion = f (l)
- Spillage flow $Q_{Spil} = f(Q_{ov}, Q_{Tu}, l)$
- Turbine head $H_{Tu} = f(Rc, l)$
- Turbine flow rate $Q_{Tu} = f(H_{Tu}, n_{Tu},$ regulating strategy)
- Turbine efficiency $\eta_{Tu} = f(H_{Tu}, Q_{Tu})$
- Power Output $P_{el} = f(H_{Tu}, Q_{Tu}, \eta_{Tu}, \eta_{Gen})$

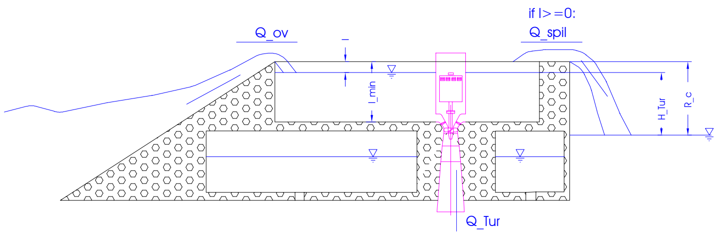


Fig. 4: Functional diagram of the Wave Dragon

By integrating the output P_{el} over a suitable period of time and comparing it to the incoming wave power, the overall efficiency of the WEC can be evaluated. This procedure has to be repeated for different sea states in order to predict the annual energy output and the average overall efficiency.

Considering the complexity of this optimisation task and the multiple interdependencies between the parameters involved, the only practical approach seemed to be a software simulation of the system behaviour combined with a systematic parameter variation. A simulation routine based on the above empirical overtopping function has been used to calculate the overtopping probability of synthetically generated wave sets, taking into account the wave distribution and the slope geometry as well as the variation of the crest height due to the immersion. Based on this routine, the whole operating process was simulated in small time steps, see [4, 5], calculating the time-dependent behaviour of all the quantities mentioned above and integrating the power output. By selective variation of the adjustable parameters, an optimum turbine configuration and operating strategy could be determined for different sea states. Finally, by applying statistical data sets for the average wave height distribution in a typical location, e.g. the Danish part of the North Sea, the parameter combination yielding the maximum annual power production could be determined.

In the following, the implications for the turbine layout resulting from the above parameter study are explained.

2.2.1. Turbine head range

In Tab. 1 a typical North Sea wave distribution is given; the waves that contribute significantly to the total energy are ranging from 1 to 5m significant height.

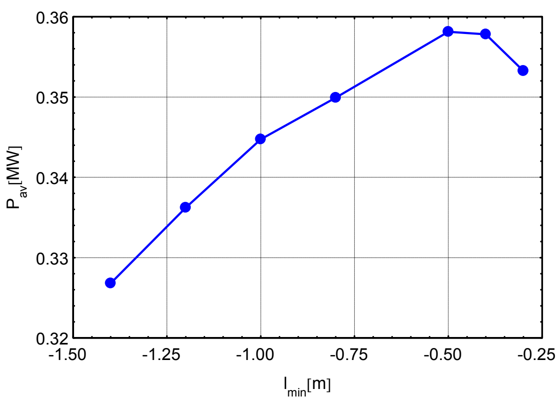
Efficient operation at different sea states requires suitable adjustment of the inlet slopes crest height. During the parameter study, the optimum crest height has been found to be closely associated with the significant wave height, while the corresponding turbine head is slightly lower due to the immersion caused by the weight of the water in the basin, see last column in tab. 1. It can be seen that the turbine has to cope with head values ranging from 0.4 to 3.0m.

H_s	Probability [%	% of total	initial crest height Rc_0	max. turbine head	min. turbine head
0	11				
1	38	6.3	0.8	0.4	0.6
2	27	22.9	1.4	0.9	1.2
3	14	30.5	2.2	1.4	1.8
4	6	25.8	2.7	2.0	2.4
5	2	14.6	3.5	2.4	3.0
>5	2				

Tab. 1: Turbine head in different sea states

2.2.2. Turbine flow range

If the turbines are only stopped when the reservoir is completely empty, some of the water volume is used at an unnecessarily low head, thus wasting a part of the potential energy. Ideally, the turbines should be stopped as soon as the reservoir is just empty enough to receive the water volume that the next wave is going to bring without any spilling. Unfortunately, neither the arrival time nor the overtopping volume of the next wave can be accurately predicted.



pFig. 5: Average power output vs. minimum basin level l_{min} at H_S=2m

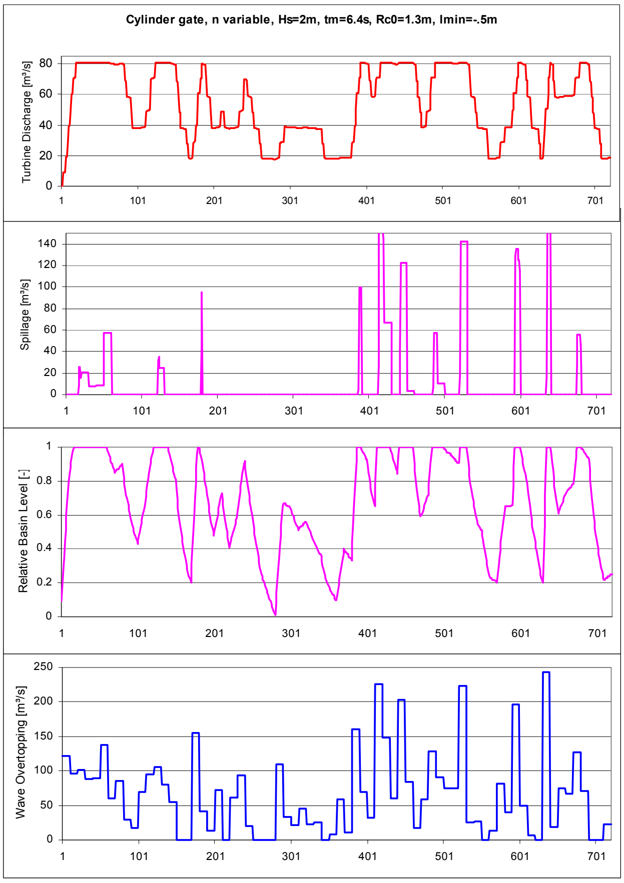


Fig 6: Time history of overtopping, basin level, spillage and turbine discharge

It has been found, however, that it is a reasonable strategy to operate the turbines at maximum flow rate when the basin is full, and decrease the flow as the basin empties until all turbines are stopped at a certain minimum basin level l_{min}. Different regulation strategies have been evaluated and the highest energy production has been obtained when the basin was only partly emptied, see Fig. 5.

In low sea states, the optimum values for the minimum level were found to be -0.4, with values up to -1.5m at high seas. In order to keep the basin level within this small range, the turbine flow has to be changed very frequently and rapidly, with relatively frequent periods of complete shutdown. Fig 6 shows a time history of the operation at a significant wave height H_S=2m. From this it can be seen that in some cases an individual turbine has to be started and stopped as often as one time per minute in order to achieve an efficient energy conversion.

3. Design of the turbines

From the above, it is apparent that the operating conditions of the turbines on the Wave Dragon differ strongly from those in a normal river hydro power station: Firstly, the turbines have to operate at very low head values ranging from 0.4m to 3.0m, which is not only on the lower limit of existing hydro power experience, but also an extremely wide variation. Secondly, due to the stochastic time distribution of the wave overtopping and the limited storage capacity, the turbines have to be regulated from zero to full load very frequently. Lastly they have to operate in a very hostile environment, with only a minimum of maintenance being possible on an unmanned offshore platform.

It was concluded that the turbines had to be as simple and rugged as possible, with an absolute minimum of moving parts. Thus, a design with both fixed guide vanes and fixed runner blades has been chosen. Efficient operation over the wide discharge range is ensured by using 16 relatively small turbines which can be switched on and off individually rather than a few large turbines. In order to grant a high efficiency throughout the wide head range, the turbines are operated at variable speed, using inverter-controlled permanent magnet generators.

Due to the low head values, it seemed advisable to use a meridional contour similar to the one of a Kaplan bulb turbine. On the other hand, the layout of the Wave Dragon dictated a vertical position of the draft tubes. Thus the water passage shown in Fig. 7 has been designed, combining a pit turbine inlet with a conical wicket gate and axial turbine runner.

During the simulations described above it has been found that for maximum energy production the turbines need to be switched on and off very frequently. In order to make this viable, two alternative solutions have been devised: A hydraulically operated cylinder gate upstream of the guide vanes and a siphon intake, see figs. 7 and 8. Both designs have been considered worth pursuing and will be tested on the small scale prototype.

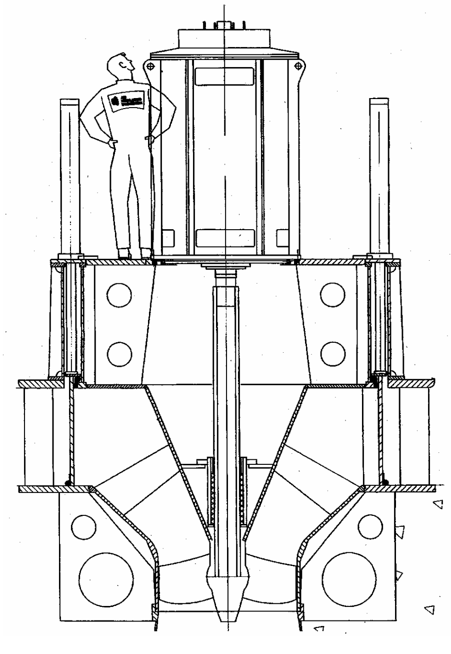


Fig. 7: Cylinder gate on/off turbine

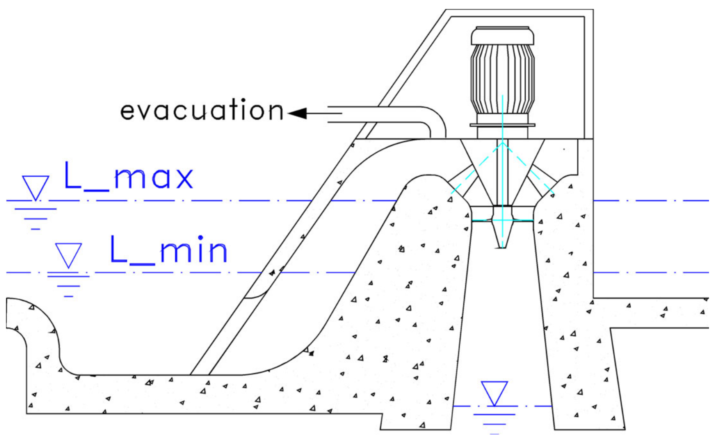


Fig 8: Siphon type on/off turbine

The turbine runner and the siphon intake structure have been designed and optimised using different CFD flow simulation tools. A model turbine has been built and tested in the Institute's laboratory.

3.1. Development of the siphon inlet duct

The advantage of the siphon solution is the absence of moving parts, which makes for a durable construction with low maintenance requirements. On the negative side, the S-shaped inlet duct implicates increased friction losses compared to the direct intake of the cylinder gate solution.

In optimising the shape of the inlet duct, the goal was to achieve low pressure losses and a uniform turbine inflow while keeping the enclosed water volume as small as possible in order to permit a quick start-up of the turbine. The latter is essential to avoid spillage losses when a big wave hits the slope of the device.

CFD flow simulations have been carried out in order to optimise the shape of the inlet duct. The calculations have been performed with the CFX-TASCflow Solver, using the Reynolds-averaged Navier-Stokes equations with the standard k-ε model on block-structured meshes.

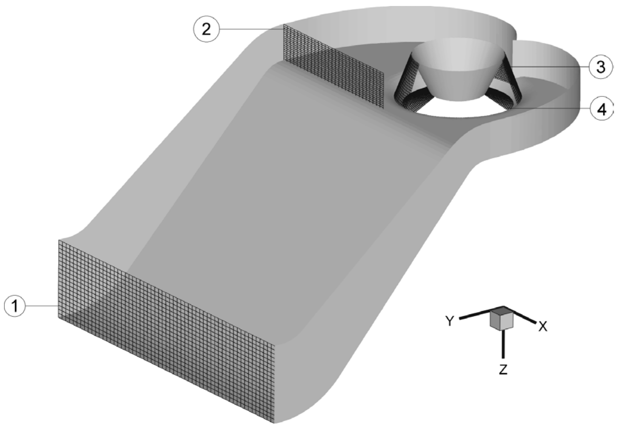
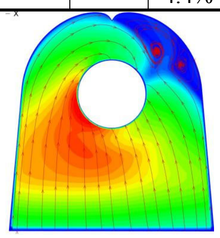
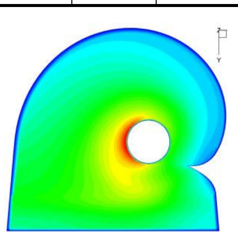
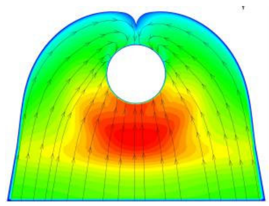
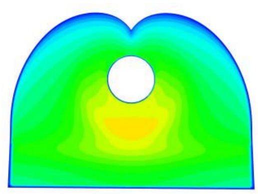


Fig. 9: 3-D view of the inlet duct with cross sections used for energy balances

A number of different configurations have been investigated. With each of the configurations, the pressure losses and the velocity distributions in the cross sections shown in Fig. 9 have been evaluated. Some of the results are displayed in Tab. 2. The first four lines are giving the minimum and maximum values of the turbine inflow velocity components and total pressure. Line 5 is giving the corresponding values of the turbine inflow angle α, the calculated relative head loss h_v/H is given in line 7.

The first configuration, labelled var. 2, was found to have an unstable flow regime in the rear part of the volute which can be seen in the upper right corner of the appropriate figure. The second configuration, which is of the semi spiral Kaplan type, showed a relatively uniform inflow, but it was found that it produced too much swirl (too small average flow angle) for a very low head turbine. In consideration of the aspects given above, the variant var. 11 was considered the optimum solution.

	var. 2			var. 4			var. 8			var. 11		
	max.	min.	Δ	max.	min.	Δ	max.	min.	Δ	max.	min.	Δ
c_{ax} [m/s]	3.2	1.6	65%	3.0	1.9	45%	3.4	1.5	78%	3.0	1.8	47%
c_u [m/s]	1.9	-1.9	3.7	3.9	2.2	1.8	1.2	-1.2	2.4	1.2	-1.2	2.4
c_{abs} [m/s]	4.2	2.8	40%	4.7	3.3	35%	4.0	1.2	108%	3.9	2.5	41%
p_{tot} [m/s]	5900	5100	15%	8600	7800	10%	8100	7800	4%	4680	4440	5%
α [°]	127°	53°	74°	65°	27°0	38°	115°	65°	50°	112°	68°	44°
$\alpha_{average}$ [°]			90°			45.0			90°			90°
h_v/H			4.4%			4.6%			7.2%			2.3%
velocity distribution in a horizontal cross section of the inlet chamber												

Tab. 2: Inflow uniformity and pressure losses for different configurations of the siphon inlet

3.2 Development of the turbine runner

For the given meridional section, a 4-bladed runner design had been previously established, which was considered a suitable solution. But as the wave tank tests indicated that there was a possibility of further improving the ramp profile in order to give more overtopping a decision was made to design a 3-bladed runner with a higher unit discharge and higher specific speed.

For the development of the runner, the flow between the guide vane inlet and the runner outlet was simulated using our in-house NS3D code with the standard k-ε model on block-structured meshes, see [6]. In order to reduce computational time, only stationary simulations with a stage interface between the guide vanes and the runner were carried out. In circumferential direction only one guide vane channel and one runner channel were considered using periodic boundary conditions. The optimisation of the runner blades was carried out by means of a real-time design system which allows to modify the blade shape of centrifugal pumps and water turbines interactively. This system, developed at the Institute, yields the flow analysis answer resulting from a geometry modification almost in real-time, depending on the level of CFD code used [7].

From the CFD simulation results, it was concluded that the 3-bladed runner had a slightly smaller efficiency at lower unit discharge values, but it holds a far greater potential for processing bigger discharges, should these become available through further development of the ramp profile. The predicted turbine efficiencies for both runners vs. unit discharge are shown in Fig. 10.

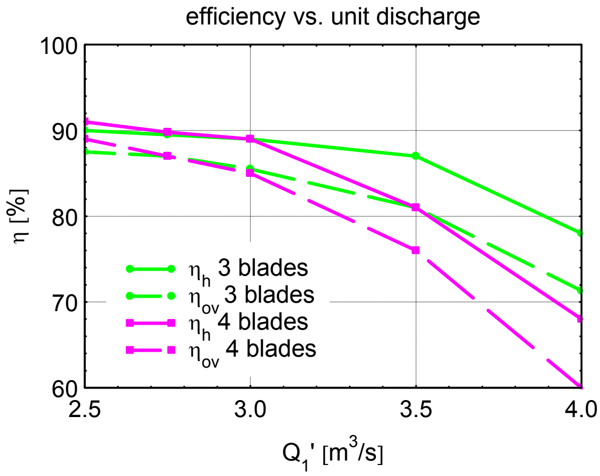


Fig. 10: Hydraulic and overall efficiency of the turbine with 3- and 4-bladed runner

3.3 Model turbine tests

Based on the above CFD simulations, a model turbine with two alternative runner configurations with 3 and 4 blades respectively, has been designed and manufactured. The turbine runners, i.e. hubs and blades, have been manufactured on a 5-axis CNC milling machine from seawater resistant aluminium bronze, as it was planned to use the model turbine on board of the sea-going small scale prototype in a later project stage.

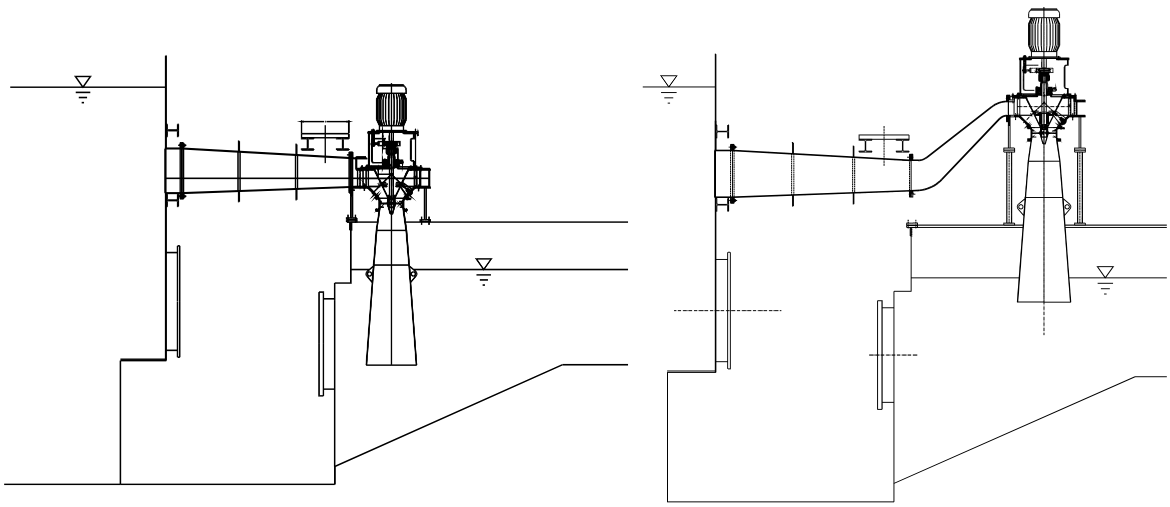


Fig. 11: Cylinder gate and siphon type model turbine in the test rig

Both configurations, cylinder gate and siphon inlet, have been investigated in an open flume low head turbine test rig. The object of the experimental investigation was to create a comprehensive data basis for choosing the most suitable turbine configuration to be used in a large-scale prototype. Fig. 11 shows the two alternative configurations in the test rig.

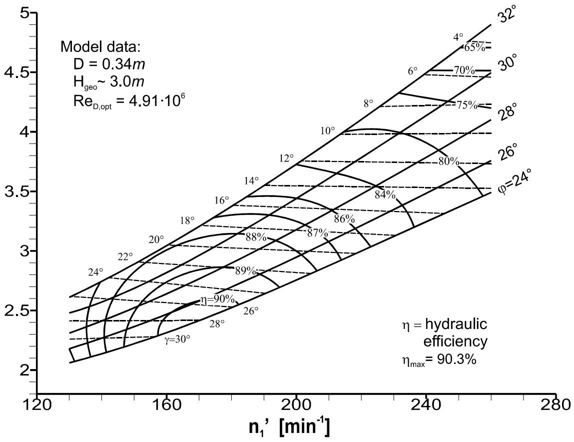


Fig. 12: Hill chart of the turbine with 3-bladed runner and cylinder gate intake

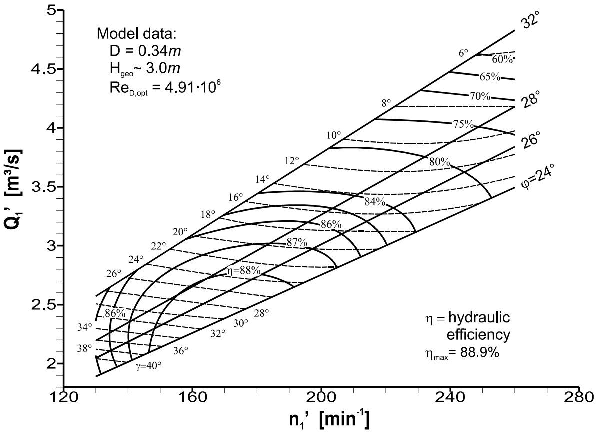


Fig. 13: Hill chart of the turbine with 3-bladed runner and siphon intake

The model tests confirmed very good performances for each of the runners in both of the inlet geometries developed. The main difference between the both runners is that the 3-bladed runner has a slightly lower peak efficiency but retains a higher efficiency at high flow rates as predicted. The peak efficiency of the turbine with the siphon intake was approx. 1.4% lower compared to the cylinder gate version, see figs 12 and 13. As the added reliability might outweigh this disadvantage it was decided that both versions will be tested on the sea-going small scale prototype.

Furthermore, model tests have been performed in order to investigate the transient process of start-up and shutdown of the siphon turbine. The object was not only to find an optimum solution granting short starting and stopping times but also to understand the fundamentals of the process in order to allow a prediction for a full scale prototype turbine to be made.

In Fig. 14, the arrangement which yielded the shortest start-up times is shown. Fig. 15 displays the quantities recorded during these tests. It can be seen that it takes about 2 seconds to fill the spiral casing (t_3-t_1). After this time, the runner is accelerated at a decreasing rate until the air has been expelled from the draft tube (t_4). Only now the torque M increases again, and the speed is raised steeply to the nominal value, which is reached after a total time of approximately 8 seconds. The shutdown time was determined to approximately 2 seconds.

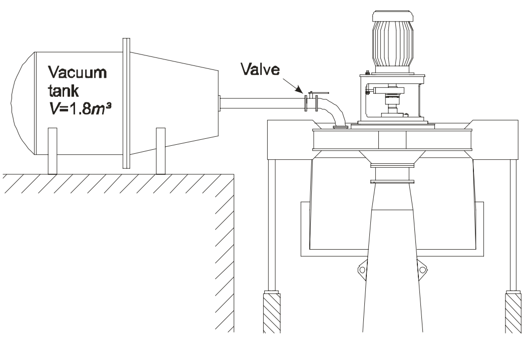


Fig. 14: Arrangement for the start-up tests

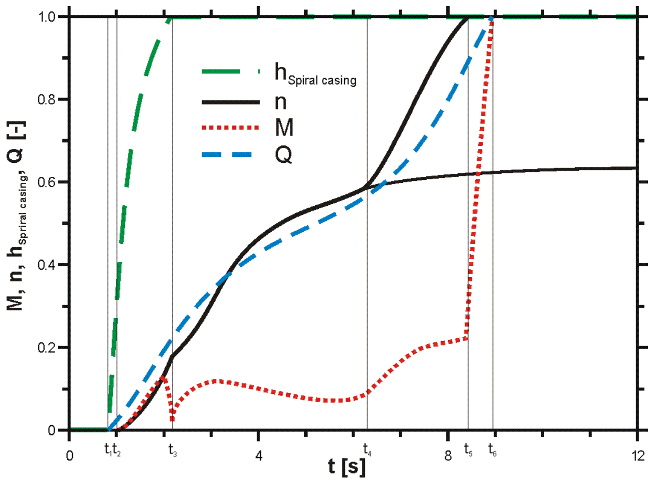


Fig. 15: Time history of the model siphon turbine start-up

4. Real sea tests on a small scale prototype, future prospects

During the first project phase, the basic design of the hull and reflector arms had been conceived and wave tank tested for overtopping as well as for structural strength in various sea conditions, including survivability tests in a 100 year storm event. Mooring and power transmission issues have been dealt with as well as ecological and economical aspects. The turbines had been designed and model tested. On the basis of these results, a 1:4.5 scale prototype has been designed, see [8].

In a second project phase, this prototype has been built. In March 2003, it has been officially deployed at the test site in Nissum Bredning, a fjord in the north of Denmark. This location had been chosen because the wave heights in this fjord are about one quarter of those to be expected in the North Sea, thus suiting the scale of the prototype.

At the present time, only the siphon type model turbine is installed on the prototype, but it will be equipped with 6 additional turbines of the cylinder gate type in September 2003. The experimental platform is equipped with all the instrumentation required on a 1:1 scale production device. In addition to this, there is a large data acquisition system tracing the readings of strain and level gauges as well as turbine related performance data. The main goals to be achieved within the course of this 3 year experimental project are:

- validation of the structural design in regular and storm operation
- verification/calibration of the overtopping formula in different sea states
- definition of optimal control strategies
- calibration of the power output simulation software

After a successful conclusion of this project phase, it will be possible to design a first generation 1:1 scale power producing Wave Dragon. The span width of the reflector arms will be about 260m, the total weight approx. 22000 tonnes, and the maximum power output will be 4 MW. A second generation Wave Dragon for the Atlantic wave climate with a yearly average of 36 kW/m wave power density will be 300m wide, have a rated power of 7 MW and produce electricity at 0.04EUR/kWh.

5. Acknowledgements

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