Numerical and experimental study of a flapping foil generator

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Abstract: The energy extraction performance of a flapping foil generator is studied through experiment and numerical simulation. A practical flapping foil generator has been proposed. The heave motion and the pitch motion of the foil are adjusted through a crankshaft-like structure. The heave and pitch motions of the foil are transferred to the rotational motion of the main shaft. A pair of gears is adopted to increase the pitch angle. A prototype with pitch amplitude $\theta_0 = 60^\circ$ has been built and the experiment is carried out in a tunnel. The overall performance of the mechanism has been analysed. Good agreement of numerical results and experiment data has been found. Further simulations with larger pitch amplitudes are carried out. It is found that higher efficiency can be achieved with larger pitch amplitude at medium frequency.

Keywords: flapping foil generator; experiment; numerical simulation; efficiency

Nomenclature

ρ density of the fluid

U speed of uniform current

C chord length of the foil

S span of the foil

l_c the distance from pivot center $X_c = (x_c, z_c)$ to the leading edge

ω circular frequency of crank arm

$T = 2\pi / \omega$ period of the oscillatory motion

$f(t)$ frequency of the crank shaft

f_a average frequency of the crank shaft

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- 26 $r(t)$ revolving velocity of the main shaft (rev/min)
- 27 r_a average revolving velocity of the main shaft
- 28 $M(t)$ torque of the main shaft
- 29 d the vertical extension of the foil
- 30 R radius of the crank arm
- 31 L length of the connecting rod
- 32 $i = r_2 / r_1$ transmission ratio of the gears, where r_1 and r_2 are the reference radius of the gears
- 33 St Strouhal number
- 34 St_a Nominal Strouhal number
- 35 $h(t)$ heave motion of the foil
- 36 $\theta_1(t)$ rotational angle of the crank arm
- 37 $\theta_2(t)$ rotational angle of the connecting rod
- 38 $\theta(t)$ pitch motion of the foil, the positive direction is defined as counter clockwise
- 39 $\alpha(t) = \arctan \frac{\dot{h}(t)}{U(t)} - \theta(t)$ the effective attack angle at the pivot
- 40 $\alpha_0 = \arctan \frac{\omega R}{U} - \theta_0$ nominal amplitude of effective attack angle
- 41 ε the phase difference of the sinusoidal heave and pitch motions
- 42 F_1, F_3, F_5 horizontal and vertical forces and moment
- 43 $f_1 = \frac{2F_1}{\rho C U^2}, f_3 = \frac{2F_3}{\rho C U^2}, f_5 = \frac{4F_5}{\rho C^2 U^2}$
- 44 $F_D = \frac{1}{NT} \int_t^{t+NT} F_1(\tau) d\tau$ average drag in N periods
- 45 $c_D = \frac{F_D}{0.5 \rho C U^2}$ drag coefficient

46 $P_{LT} = F_3(t)\dot{h}(t)$ the instantaneous power due to heave motion and $c_{PLT} = \frac{P_{LT}}{0.5\rho CU^3}$

47 $P_{MT} = F_5(t)\dot{\theta}(t)$ the instantaneous power due to pitch motion and $c_{PMT} = \frac{P_{MT}}{0.5\rho CU^3}$

48 $P_{cal}(t) = P_{LT}(t) + P_{MT}(t)$ the instantaneous power of the foil

49 $P_L = \frac{1}{T} \int_t^{t+T} F_3(\tau)\dot{h}(\tau)d\tau$ average power due to heave motion and $c_{PL} = \frac{P_L}{0.5\rho CU^3}$

50 $P_M = \frac{1}{T} \int_t^{t+T} F_5(\tau)\dot{\theta}(\tau)d\tau$ average power due to pitch motion and $c_{PM} = \frac{P_M}{0.5\rho CU^3}$

51 $P_{cal} = P_{LT} + P_{MT}$ average power of the foil and $c_P = \frac{P_{cal}}{0.5\rho CU^3}$

52 $\eta_{cal} = \frac{P_{cal}}{0.5\rho U^3 d}$ the calculated efficiency

53 P_{test} tested average power of the generator from shaft

54 P_{ML} mechanical losses of the generator

55 P_{exp} experimental average power of the foil

56 $\eta_{exp} = \frac{P_{exp}}{0.5\rho U^3 d}$ the experimental efficiency

57 1. Introduction

58 Extracting kinematic energy from wind and tidal current is a promising supplement of the fossil
 59 energy^[1]. Compared with the vertical and horizontal axis turbines^[2], flapping foil is a relatively new
 60 version of mechanism to extract kinematic energy of tidal current. The flapping foil, which originates from
 61 the biomimetic propulsion, has a lower translational velocity; it will bring less risk to animals^{[3][4]}.
 62 Although the flapping foil generator is less efficient than the horizontal axis turbines (up to 45%^[5]), it can
 63 operate in a relatively low speed current^[3]. The flapping foil has a rectangular swept area; when tidal
 64 energy in shallow water is concerned, it would extract more power than horizontal axis turbine as the
 65 span of the foil can be extended horizontally. In the perspective of long-term operation on the sea, the
 66 turbine has to envisage the variation of current speed and waves; easy maintenance and the reliability of

the mechanical system will be the issues. A robust and reliable energy converter is required.

Energy harvesting of flapping foil had been investigated through numerical simulations. The optimum parameters for higher efficiency had been studied. As the foil performs the heave and the pitch motions, the effective attack angle of the foil and the incoming flow shall be positive during upstroke and be negative during downstroke^[6]. To describe the sinusoidal motion of the oscillatory foil, Strouhal number, heave amplitude, pitch amplitude and the phase difference, which are denoted as St , h_0 , θ_0 , ε , are introduced to determine the trajectory. A number of researches^[7-12] have studied the effects of these parameters on the efficiency. It is expected that the efficiency can be generally enhanced by increasing the effective attack angle. Inspired by the non-sinusoidal motion for propulsion^[13], non-sinusoidal heave and pitch motions^[14-16] had been introduced to improve the efficiency. The proposed trajectories have actually increased the mean effective attack angle during the upstroke and the downstroke; the efficiency can be improved except that severe leading edge vortex shedding occurs. However these idealized trajectories are not practical in the stage of the mechanical design.

In the perspective of application, efficient and reliable mechanical design is one of the key issues of tidal current energy extraction. There are practical difficulties in the mechanical design for those prescribed sinusoidal or non-sinusoidal flapping motions. A few demonstrations or prototypes have been built to study the energy harvesting through flapping foil. The pioneer work was the 'wind mill' proposed by McKinney and DeLaurier^[17]. After that, Engineering Business Limited developed 'Stingray'^[18-20]. However, this semi-passive convertor is estimated with a relatively low efficiency, because of its insufficient pitch amplitude^[4]. Huxham et al^[21] had improved the mechanism of 'stingray'; pitch amplitude with 72° was adopted. The highest efficiency of the turbine reached as high as 27%. Pulse Tidal^[22] had constructed a demonstration with bi-direction flapping foil. Jones et al^[23] carried out experimental study of two foils in tandem configuration; the efficiency was much lower than the theoretical results. The discrepancy could be due to mechanical losses. The experimental study of tandem-foils turbine had been conducted by Kinsey et al^[24]. The efficiency of one single foil is around 20% and the total efficiency of the two-foil system reached 40% when 10% mechanical loss is excluded. It appears that a novel design with low mechanical losses is most valuable.

A new version of flapping foil generator is proposed in present study. The performance of the

proposed tidal energy generator will be investigated numerically and experimentally. The mechanism transfers the heave and pitch motions of the foil into the rotary motion of the shaft through a crankshaft-like structure. To increase the effective attack angle, a pair of gears is adopted. A prototype of the generator has been built and the experiment is carried out in a tunnel. The generator is arranged above the water surface. It would cost less as the water proof and seal problem have been avoided. The maintenance would be convenient. The experiment is carried out with various oscillating frequencies and different current speeds. The power of the turbine is tested at the shaft. The mechanical losses are estimated. The total power would be the tested power at the shaft plus the mechanical losses. In the perspective of energy conservation, the extracted power of the foil is equal with the total power measured at the shaft. Numerical simulations of pitch amplitude with $\theta_0 = 60^\circ$ are carried out firstly; the angular velocity of the shaft is constant. The energy extraction performance at various Strouhal number is analysed. After the numerical results are validated, further simulations with pitch amplitudes $\theta_0 = 65^\circ$, $\theta_0 = 70^\circ$ and $\theta_0 = 75^\circ$ have been performed. The efficiency of the mechanism is analysed and the most preferable frequency has been found.

2. Description of the flapping foil generator

Fig.1 The concept design of the flapping foil generator, (a) illustration of the mechanism (b) overview (c) gears and the rail.

The concept design of the flapping foil generator is shown in Fig.1. The foil will be pushed moving up and down along the rail vertically when it is subjected to tidal current. Here the heave motion refers to the translational motion that is perpendicular to the current. The crankshaft is driven to rotate clockwise. Lead masses, which perform the role of the flying wheel, are fixed at the middle of the shaft. Its inertia can keep the rotational motion as the crank arm passing the highest or lowest position. A pair of gears, as illustrated in Figs.(1c), is adopted to increase the pitch angle. The transmission ratio of the gears is defined as $i = r_2 / r_1$, where r_1 and r_2 are the reference radius of the gears. The heave motion is determined by the crank arm with a length of R and the connecting rod whose length is L . The rotational angle of

the crank arm is denoted as θ_1 . The coordinate system oxz is fixed at the center of the main shaft.

Taking θ_1 as the reference, we have the heave motion and rotational motion of the foil

$$h(t) = R \sin \theta_1 - L \cos \theta_2 - r_1 - r_2 = R \sin \theta_1 - L \cos(\arcsin \frac{-R \cos \theta_1}{L}) - r_1 - r_2 \quad (1)$$

$$\theta(t) = -i\theta_2 = i \arcsin \frac{R \cos \theta_1}{L} \quad (2)$$

A prototype has been built as illustrated in Fig.1. Some parts are made from stainless steel or bakelite plate. A NACA0015 foil with chord length $c = 8.0cm$ is adopted. The span of the foil $S = 64.0cm$. Two end plates are fixed at the ends of the foil to reduce the three-dimensional effects. The crank arm $R = 8.0cm$ and the length of the connecting rod $L = 43.9cm$. The transmission ratio of the gears $i = 5.714$. Hence we have the nominal pitch motion amplitude $\theta_0 = 60^\circ$ when $\theta_1 = 0$. Two lead masses with $6.8kg$ are fixed on the main shaft. The mechanism can rotate smoothly as it is started by an external force. The shaft is connected with the torque meter and the generator. A slide rheostat, with its maximum electric resistance 24Ω , is connected in the circuit of the generator. The general parameters of the flapping foil are shown in table 1.

Table 1 Parameters of the flapping foil generator

Foil section	NACA 0015
Chord length C	8.0cm
Span S	64.0cm
Pitching pivot l_c	2.67cm
Heaving amplitude h_0	8.0cm
Pitching amplitude θ_0	60°

Fig.2 The prototype fixed in the tunnel

The experiment is carried out in a tunnel in Harbin Engineering University. The current in the tunnel is generated by the main impeller driven by a 40KW direct current electromotor; guide vanes and grids are utilized to make a uniform current. The current velocity U ranges from $0 \sim 2.0m/s$. The length of

the working area is approximately $7m$. The working area has a rectangular cross section with $1.70m \times 1.50m$ (wide $\times$ depth).

The flapping foil generator is fixed in the middle of the working area of the tunnel by four fixtures, as shown in Fig.2. The torque meter and the generator are connected with the crankshaft through crosshead couplings. The torque meter is calibrated before the experiment. The output signal of torque meter includes both torque and angular velocity. The signal is processed by the data collection card. The error of instantaneous torque is within 0.02% and the error of the angular velocity is within 0.05%. When the foil is at its top position, the distance of the chord center to the water surface is 19cm, which is approximately 2.4 times of the chord length.

3. Experiment and data analysis

Fig.3 The time histories of (a)(c)&(e) revolving velocity and (b)(d)&(f) the torque of the shaft at

$$U = 0.98m/s.$$

The experiment is carried out with various current speeds. The instantaneous revolving velocity of the shaft $r(t)$ and the corresponding torque $M(t)$ are recorded. Fig.3 shows the typical time history of the revolving velocity $r(t)$ and torque of the main shaft $M(t)$ at $U = 0.98m/s$. The raw data is filtered using fast Fourier transform (FFT) algorithm. As shown in the figure, $r(t)$ and $M(t)$ are always positive but with small amplitude fluctuations. The average frequency of the shaft $f(t)$ in N oscillating periods is estimated as follow

$$f_a = \frac{1}{NT} \int_t^{t+NT} f(t)dt = \frac{1}{60NT} \int_t^{t+NT} r(t)dt \quad (3)$$

and the nominal Strouhal number is defined as

$$St_a = \frac{2f_a R}{U} \quad (4)$$

The average power of the mechanism in N oscillating periods can be obtained

$$P_{test} = \frac{1}{NT} \int_t^{t+NT} M(t) \dot{\theta}(t) dt = \frac{\pi}{30NT} \int_t^{t+NT} M(t) f(t) dt \quad (5)$$

Fig.4 The mechanical loss against angular velocity

Considering the dynamic mechanical friction, the mechanical losses of the system must be excluded. The power computed from Eq.(5) equals the total power extracted from the current minus the mechanical losses P_{ML} . To estimate this mechanical losses, the power required to drive the generator without hydrofoil at different angular frequencies is taken as the mechanical losses. Fig.4 shows that the required power to drive the mechanical system is less than 0.7 Watt; it is much lower than the extracted power which will be presented below. The third order polynomial fitting is used to approximate the mechanical losses. The mechanical loss P_{ML} at any given angular velocity can be obtained through interpolation. We have

$$P_{ML} = 0.000752 + 0.00272r - 0.00004173r^2 + 0.000000305r^3 \quad (6)$$

Comparing the test with hydrofoil in the tunnel, the tested mechanical losses of the bearings and gears in the air without hydrofoil will be minimal. Therefore, the mechanical friction losses of the system would be underestimated. The estimated P_{ML} is considered as a minimal bound for the actual mechanical losses.

The corresponding power and efficiency can be calculated as follow

$$P_{exp} = P_{test} + P_{ML} \quad (7)$$

$$\eta_{exp} = \frac{P_{exp}}{0.5\rho d U^3} \quad (8)$$

where d is the maximum vertical extent of the flapping foil.

Fig.5 (a) and (b) the average power against revolving velocity and Strouhal number respectively, (c)

efficiency at various current speeds.

Tests with different angular velocities are carried out as we change the electric resistance of slide-wire rheostat. Fig.5 shows the average power and the efficiency of the prototype at different current speeds and various frequencies. At lower revolving velocity or smaller St_a , the power of the prototype increases; it reaches its peak when St_a is around 0.2 and decreases as St_a keeps increasing. When the same St_a is considered, the power increases as the current speed. However, the curves of efficiency at different current speeds are very close. It can be seen that the effects of current speed have less influence on the efficiency of the generator. The highest efficiency is around 0.26 when $St = 0.2$.

Table 2 The nominal scale of the mechanical losses at various revolving speeds

	$U = 0.77$	$U = 0.88$	$U = 0.98$	$U = 1.08$	$U = 1.19$
r_a	63.04	68.65	73.23	82.36	93.5
$P_{ML} / P_{\max} (\%)$	1.15	0.83	0.65	0.58	0.55

We note in Fig.(5a) that the mechanical loss P_{ML} is much lower than the extracted power. The scale of the losses will increase since P_{ML} increases and the extracted power decreases as the revolving velocity keeps increasing. The ratio of the P_{ML} over the maximum power $P_{\max}$ is taken as the nominal scale of the mechanical losses. As shown in table 2, the losses are lower than 1.2%. It decreases as the current speed increases. Data in the table suggests that present model-scale mechanism is high efficient in mechanical transmission.

4. Numerical simulations and performance analysis of the flapping foil generator

The proposed mechanical system is composed of crank shafts, gears and the foil. The experimental data show that the overall system operates at a relatively steady state. The angular velocity and torque of the shaft fluctuate around their mean value due to the dynamic couplings. To perform the numerical simulation of the foil in the current, it will bring great convenience when the average angular velocity of the shaft is adopted to describe the motion of the shaft. In numerical simulation, the angular velocity of

the crank shaft is constant or $\theta_1 = \omega t$. The angular motion of the connecting rod can be written as

$$\theta_2(t) = \arcsin \frac{-R \cos \omega t}{L} \quad (9)$$

$$\dot{\theta}_2(t) = \frac{R \omega \sin \omega t}{\sqrt{L^2 - R^2 (\cos \omega t)^2}} \quad (10)$$

The heave motion of the foil becomes

$$h(t) = R \sin \omega t - L \cos(\arcsin \frac{-R \cos \omega t}{L}) - r_1 - r_2 \quad (11)$$

$$\dot{h}(t) = \omega R \cos \omega t - \frac{\omega R^2 \sin 2\omega t}{2\sqrt{L^2 - R^2 (\cos \omega t)^2}} \quad (12)$$

The corresponding pitch motion of the foil can be written as

$$\theta(t) = -i \theta_2 = i \arcsin \frac{R \cos \omega t}{L} \quad (13)$$

$$\dot{\theta}(t) = -i \dot{\theta}_2(t) = \frac{-i \omega R \sin \omega t}{\sqrt{L^2 - R^2 (\cos \omega t)^2}} \quad (14)$$

The hydrodynamic forces are obtained through pressure integration. We have the calculated average power

$$P_{cal} = P_{LT} + P_{MT} \quad (15)$$

and

$$c_p = \frac{P_{cal}}{0.5 \rho C U^3} \quad (16)$$

where $P_L = \frac{1}{T} \int_t^{t+T} F_3(\tau) \dot{h}(\tau) d\tau$ and $P_M = \frac{1}{T} \int_t^{t+T} F_5(\tau) \dot{\theta}(\tau) d\tau$ are the average power due to vertical force and the moment respectively.

The efficiency based on numerical simulation can be obtained

$$\eta_{cal} = \frac{P_{cal}}{0.5 \rho U^3 d} \quad (17)$$

In the perspective of energy conservation, the power coefficient and the efficiency of the foil obtained through Eqs.(16) and (17) will be the same as the tested results at the shaft.

Fig. 6 Computational domain and mesh details of the flapping foil.

Simulations are conducted using the commercial CFD code Fluent (version 14.5). The dynamic meshes technique and the sliding meshes are used to implement the oscillatory motion of the foil. Fig.6 shows the computational domain and details of the mesh. The non-slip condition on the foil surface is adopted. The velocity inlet is prescribed and the outflow condition has been used. The near body resolution on the foil surface satisfies $y^+ \approx 1$. The present simulations use a two-dimensional URANS (unsteady Reynolds-average Navier-Stokes) approach with the one-equation Spalart-Allmaras turbulence model (strain/vorticity based production). The PISO algorithm is chosen for pressure-velocity coupling. Second order schemes are used for momentum and turbulent viscosity discretizations. PRESTO! scheme is used for pressure discretization. First order implicit scheme is selected for the unsteady formulation. Absolute convergence criteria of 10^{-5} is adopted for all residuals.

Tests of mesh size and time resolution independence are conducted firstly. The sinusoidal heave motion $h(t) = h_0 \sin \omega t$ and pitch motion $\theta(t) = \theta_0 \sin(\omega t + \pi/2)$ with $St = 0.28$, $\theta_0 = 75^\circ$, $h_0 = 1.0$ are considered. The chord length $C = 24cm$ and the inlet free stream velocity $U = 2.08m/s$ are adopted. Two mesh densities are used. The low density mesh (mesh a) has 75 000 cells with 280 grids on the foil surface; the high density mesh (mesh b) has 140 000 cells with 380 grids on the foil surface. Two temporal resolutions are chosen as $\Delta t_1 = T/1000$ and $\Delta t_2 = T/2000$. Fig.7 shows the time history in the 6th oscillation. The curves of forces f_1 , f_3 and f_5 are in good agreement, though small discrepancies of f_3 and f_5 can be observed when $t \approx 0.05$ and $t \approx 0.55$. The overall performance of the flapping foil is not affected significantly. As shown in table 3, the power coefficients c_{PL} , c_{PM} and η_{cal} are in good agreement. The energy harvesting efficiency is not sensitive to the mesh density and the time resolution. The results of mesh b with Δt_1 are further compared with the numerical results of Kinsey& Dumas^[25]. As shown in table 4, c_D , the maximum f_3 and f_5 and the power coefficient c_p are in good agreement.

Fig.7 The time histories with different meshes and temporal resolution, (a) f_1 (b) f_3 (c) f_5

Table 3 The performance of the flapping foil with different mesh densities and time resolutions.

	Mesh a ($\Delta t_1=T/1000$)	Mesh a ($\Delta t_2=T/2000$)	Mesh b ($\Delta t_1=T/1000$)	Mesh b ($\Delta t_2=T/2000$)
c_{PL}	1.127	1.110	1.164	1.174
c_{PM}	-0.1884	-0.1783	-0.2113	-0.2215
$\eta_{cal}(\%)$	36.78	36.52	37.36	37.35

Table 4 Comparison of forces and power coefficient with the results of Kinsey& Dumas^[25]

	c_D	Maximum f_3	Maximum f_5	c_P	$\eta_{cal}(\%)$
Present (mesh b, Δt_1)	1.755	2.685	1.144	0.953	37.36
Kinsey& Dumas^[25]	1.757	2.819	1.130	0.986	38.68

A two-dimensional NACA0015 foil with same paramters in table 1 at various oscillating frequencies is simulated. The oscillatory motions are prescribed in Eqs.(11)-(14). The inlet free stream velocity $U = 1.0m/s$ and the corresponding Reynolds number $Re = 80000$. The mesh with 140 000 cells with 380 grids on the hydrofoil surface and time resolution $\Delta t_1 = T/1000$ are adopted. The efficiency η of the flapping foil generator is shown in Fig.8. Comparing with the experimental data, good agreement has been found.

Fig.8 The efficiency of the foil at various frequencies with $\theta_0 = 60^\circ$.

The typical time histories of the hydrodynamic force and the power due to heave and pitch motions at $St = 0.15$, $St = 0.2$ and $St = 0.25$ are shown in Fig.9. It can be found that the horizontal force amplitude is larger when $St = 0.15$; while the vertical forces at $St = 0.15$, $St = 0.2$ and $St = 0.25$ have similar amplitude. For the case of $St = 0.15$, sharp variations of curves of f_1 , f_3 and f_5 can be found when $t/T \approx 0.1$ and $t/T \approx 0.6$. For the case of $St = 0.25$, the force curves are smooth. Comparing Figs.(9b), (9d) and (9f), the curves of c_{PLT} have larger amplitude as St increases; the curve c_{PMT} in Fig.(9b) is above or close to zero line; while c_{PMT} in Fig.(9f) is always negative. This indicates

that the pitch motion consumes energy when $St = 0.25$.

Fig.9 The time history of the forces and power due to heave and pith motion, (a)(b) $St = 0.15$ (c)(d) $St = 0.20$ (e)(f) $St = 0.25$.

In Fig.10, snapshots of the flow field give further insight of the problem. When $St = 0.15$, the leading edge vortex shedding (LEV) is obvious when the foil approaches the top position. The LEV would cause the reduction of the vertical force and moment; as a result, it would cause the reduction of the power. When $St = 0.2$, the LEV is delayed. The power would not be reduced significantly. When $St = 0.25$ there is no LEV. The occurrence of leading edge vortex shedding shall be related to the effective attack angle. As shown in table 5, the amplitude of the effective attack angle α_0 is too large when $St \leq 0.2$; the efficiency would be reduced because of the LEV. The peak efficiency is achieved when severe LEV is reduced as St increases. As St increases further, α_0 decreases significantly. Consequently, the efficiency drops due to the insufficient effective attack angle.

Table 5 The nominal effective attack angle amplitude at various St

St	0.1	0.15	0.2	0.25	0.3	0.35	0.4
α_0 (degree)	42.4	34.6	27.7	21.7	16.6	12.1	8.4

Fig.10 The snapshots of the flow around the foil at typical positions

Further simulations are carried out with larger pitch amplitudes. The performance of the generator is shown in Fig.11. The power due to vertical force c_{PL} increases as St is small; it decreases or has a decreased ratio when St increases further. It can be seen in Fig.(11a) that c_{PL} with larger θ_0 is much higher than those with smaller θ_0 when $St > 0.2$. In Fig.(11b), the power due to pitch motion is positive at smaller St ; it becomes negative as St increases. Consequently, the power coefficients increase at smaller St ; the peaks of efficiency will be found at medium St . We notice that the efficiency with $\theta_0 = 70^\circ$ and $\theta_0 = 75^\circ$ is over 0.3 when $St = 0.2 \sim 0.35$, indicating that high performance exists within a relatively wide range of frequency. The efficiency of larger θ_0 is higher than those of

smaller pitch amplitudes when $St > 0.2$; however the increment of the efficiency from $\theta_0 = 70^\circ$ to $\theta_0 = 75^\circ$ is less than those from $\theta_0 = 60^\circ$ to $\theta_0 = 65^\circ$ and from $\theta_0 = 65^\circ$ to $\theta_0 = 70^\circ$. The peaks of the efficiency appear at larger St as θ_0 increases. The highest efficiency is 34% when $\theta_0 = 75^\circ$ and $St = 0.275$.

Fig.11 The performance of the flapping foil generator with various pitch amplitudes.

Concluding remarks

The performance of a practical flapping foil generator has been investigated experimentally and numerically. The heave and pitch motions of the foil are transmitted into rotational motion of the shaft through crankshaft structures. Although the trajectory of the foil is confined by the crankshaft and the gears, the proposed design has low mechanical losses. The efficiency of present generator is higher than most existing single flapping foil turbines. Although the efficiency of single foil turbine would be lower than the rotary turbine. The tandem foils based on present design can be adopted to increase the overall efficiency.

Numerical results find good agreement with experimental data. Simulations with $\theta_0 = 60^\circ$, $\theta_0 = 65^\circ$, $\theta_0 = 70^\circ$ and $\theta_0 = 75^\circ$ show that the efficiency increases significantly as θ_0 increases. When St is small, the effective attack angle is too large. The occurrence of LEV will cause significant reduction of power coefficient. As St increases, the effective attack angle decreases; the LEV will be delayed or even disappear. The peak efficiency would be achieved. As St increases further the efficiency will decrease due to insufficient effective attack angle. The efficiency is over 30% when $St = 0.2 \sim 0.35$ with $\theta_0 = 70^\circ$ or $\theta_0 = 75^\circ$. The generator has higher efficiency with a relatively wide frequency when the pitch amplitude is large enough. When the current speed decreases, the motion frequency will decrease but the efficiency would remain the same level as long as $St = 0.25 \sim 0.35$. To scale the present model-scale turbine to a full-scale design, the motion of the turbine could share the same St , which is a non-dimensional frequency. Other parameters, including R/C , L/C and $i = r_1/r_2$, can adopt those of present model. The suggested operation conditions would be $St = 0.2 \sim 0.35$, $\theta_0 = 70^\circ \sim 75^\circ$. The full

346 scale turbine has to envisage the sea environment; the gears and the rail shall be replaced by robust
347 components.

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