An Experimental Study of a Bidirectional Radial Turbine for Pneumatic Wave Energy Conversion

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Abstract - Results of a wind tunnel studies of a bidirectional radial turbine designed for pneumatic water wave energy conversion are presented. The turbine is a single-stage type having one row of rotor blades and two rows of stator blades. The test results compare well with those obtained from a test of a counter-rotating bidirectional axial turbine tested under the same conditions. Because of the relatively simple design of the radial turbine, its manufacture is less complex, making the radial turbine more cost-effective. Both turbines are designed to convert wave-induced air flow energy into electrical energy.

1. INTRODUCTION

Since the mid 1960's, the conversion of the energy of water waves into electrical energy has become a reality, primarily due to the efforts of Yoshio Masuda of Japan. Most of the early efforts of Masuda and others are described in [1]. Until the early 1980's, Masuda's efforts were directed toward pneumatic wave energy conversion. This technology can be understood by referring to the Fig. 1, where a vertical capture chamber having an open bottom in the water and a turbine passage over the top opening is sketched. As waves pass the capture chamber, a free communication occurs between the ambient and internal waters. Hence, the internal water column oscillates at the frequency of the passing wave, and acts as a piston, alternatingly driving air above its free-surface out of the turbine passage and drawing air into the passage. The alternating air flow, in turn, excites a turbo-electric system mounted in the passage.

The turbines used in pneumatic wave energy conversion systems fall into two general categories. These are unidirectional and bidirectional. The first turbines used in wave energy projects where unidirectional, requiring flaps to rectify the air flow past the turbine, as illustrated in Fig. 2. The operation and performance of these turbine are discussed in [2], [3] and [4]. Because of the possibility of flap fouling (causing the flap to remain either open or closed), several bidirectional turbines were designed. These bidirectional turbines include those of Wells [5], Babinsten [6], Filipenco [7] and McCormick [8], [9]. These turbines are all axial flow turbines. The McCormick counter-rotating turbine is that sketched in Fig. 1. The turbine discussed herein is radial in design, having a single rotor with arcuate blades and two rows of stators or guide vanes, as sketched in Fig. 3. Because of the design, the rotor travels in the same direction regardless of the direction of the air flow.

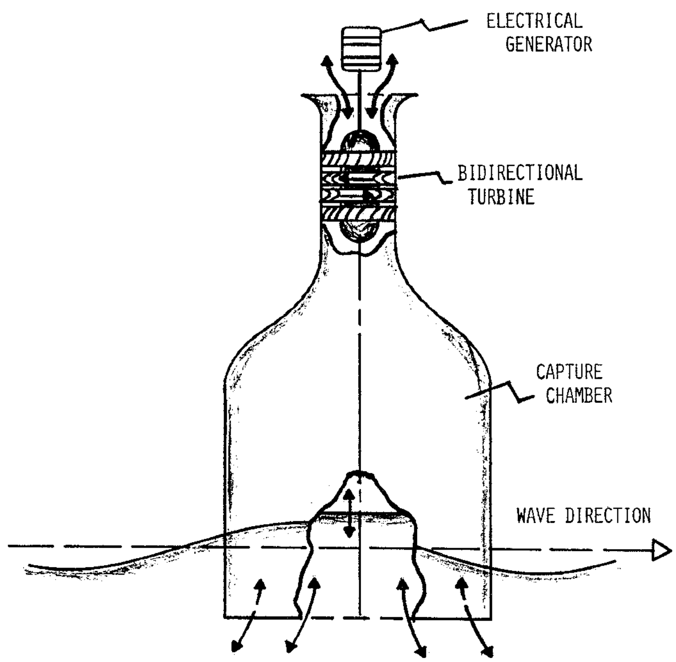

Fig. 1. Sketch of a Vertical Pneumatic Wave Energy Conversion System with a Bidirectional Counter-Rotating Turbine.

2. TURBO-GENERATOR SYSTEM MODEL STUDY

The turbo-generating system studied consists of the radial turbine (Fig. 3), a set of pulley drives, a dc generator and a rheostat. Referring to Fig. 4, the rheostat, in series with the generator, provides a an adjustable impedance. Prior to testing the complete system on the table-top wind tunnel in the U. S. Naval Academy's Aerospace Laboratory, tests were conducted using a frictional braking system on the turbine drive shaft (Fig. 5) so that the performance of the turbine alone could be evaluated. This same system was used on a bidirectional axial flow turbine of the same scale in the study leading to [9]. Hence, a direct comparison of the performances of the radial and axial turbines can be made. Following the frictional loading tests, the complete radial turbo-generator system was tested using the same wind tunnel. The exhaust of the wind tunnel is in the plane of the table top. Because of the this, both the radial and axial turbines were mounted with their axes vertical.

3. TEST RESULTS

Results of the tests with the frictional breaking system are presented in Figs. 6 and 7. In Fig. 6 the maximum power generated by both the radial and counter-rotating axial turbines is presented as a function of the total pressure in the plenum chamber. The total pressure is used so that a direct comparison of the two turbines can be made. The tests on the radial turbine are from two separate studies. The reader can see that for a total pressure of approximately 100 N/m^2 the power generated by the turbines to overcome the frictional load is approximately the same, i. e. a little less than 5 Watts. As the total pressure increases, the radial turbine appears to out-perform the axial turbine, producing nearly twice the power at approximately 500 N/m^2 total pressure. The power in this study is simply the product of the frictional torque and the rotational speed of the turbine. That is

$$P = T\omega = Wr\omega \qquad (1)$$

where W is the applied force at a distance r from the axis of rotation, and where ω is the rotational velocity.

In Fig. 7, the performance curves (at a pressure of approximately 100 N/m^2) for both the radial and bidirectional turbines are presented, where the power required to overcome the frictional load is shown as a function of the rotational speed of the turbine. One sees that the peak power output of the counter-rotating

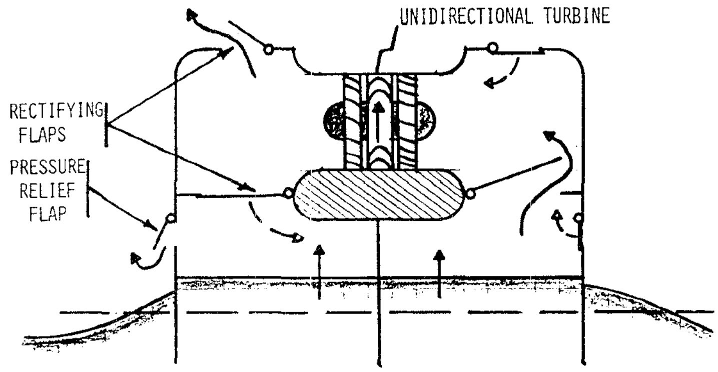

Fig. 2. Pneumatic System with a Unidirectional Turbine and Rectifying Flaps.

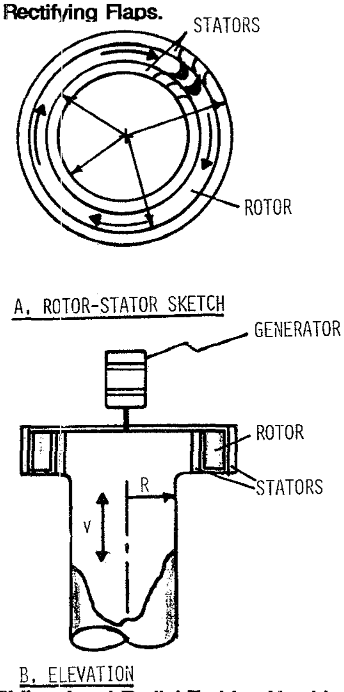

Fig. 3. Sketch of the Bidirectional Radial Turbine Used in the Present Study.

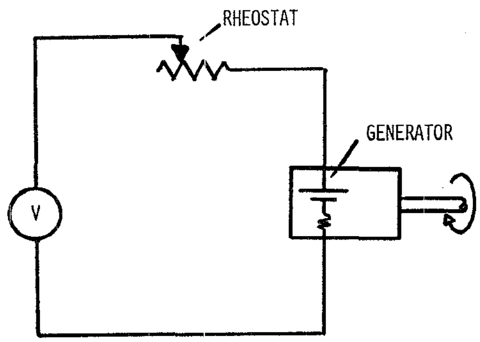

Fig. 4. Circuit Diagram for the Radial Turbine Wind Tunnel Tests.

turbine is approximately 10% higher than that of the radial turbine. The performance curve of the radial turbine is broader, having a runaway speed that is approximately 25% higher than that of the radial turbine.

The final friction tests were designed to obtain operational curves for a number of total pressures and corresponding air velocities. Results of those tests are presented in Fig. 8. where the power to overcome the frictional load is presented as a function of the turbine rotational speed for three air velocities.

The next series of wind tunnel tests were conducted with the electrical system connected to the radial flow turbine. A series of gears was required to step up the rotational speed of the turbine to match the required speed of the dc generator. Specifically, the overall setup ratio was 6.25. The first of these tests was designed to determine the effect of the variable electrical resistance in the circuit sketched in Fig. 4. Results of these tests are presented in Fig. 8, where the energy conversion efficiency is presented as a function of rotational speed for six electrical resistance values. For all rotational speeds the 22Ω resistance results in the highest conversion efficiency. This is not necessarily the impedance-matched resistance, since the resistances in the neighborhood of this value were not studied. The actual power developed during these tests is shown in Fig. 9, where the electrical power is shown as a function of air speed for three impedance values, including 22Ω. As can be seen in that figure, there are small-scale losses in the system which become less significant as the size of the system increases.

As in the study leading to [9], an attempt was made to study the radial turbine in the U. S. Naval Academy's 117-meter wave and towing tank. Unfortunately, because of a malfunction in the wavemaker, the required wave heights for given wave frequencies could not be obtained. Hence, for this study, only the wind tunnel results are available.

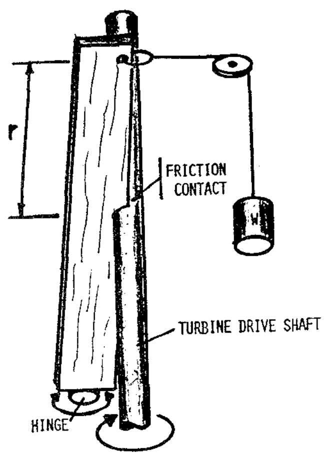

Fig. 5. Friction-Torque Device.

4. DISCUSSION AND CONCLUSIONS

Results of wind tunnel tests of two bidirectional turbines designed for pneumatic wave energy conversion are presented. The designs of the turbines are radically different, one being axial flow while the other being radial flow. The radial flow turbine, which is the subject of the present study, appears to have a significantly better performance than the axial turbine at higher operational pressures (between 200 and 500 N/m^2), as seen in Fig. 6. At a total pressure of approximately 100 N/m^2 the peak power values of the two turbines are within 10% of each other, as seen in Fig. 7. The performance curve of the radial flow turbine, however, is broader, having a runaway speed that is about 25% higher than that of the axial flow turbine.

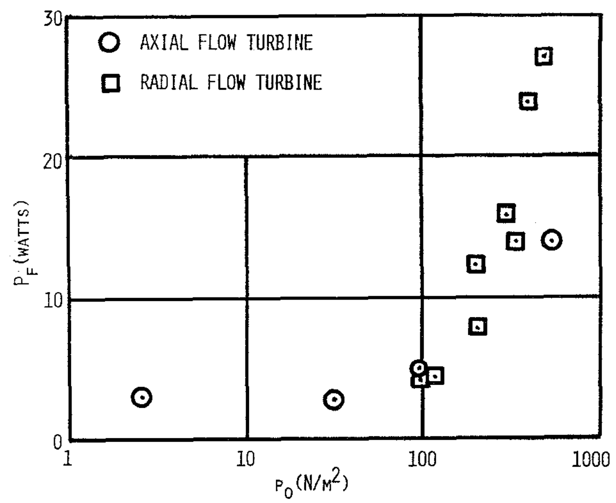

Fig. 6. Peak Turbine Powers in the Frictional Loading Tests.

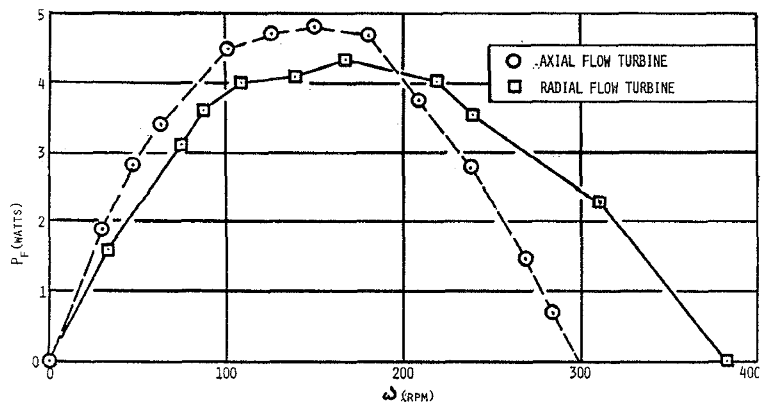

Fig. 7. Turbine Performance in the Frictional Loading Tests.

Since controlled wave tank tests of the radial flow turbine were not possible, the impedance-matched performance curve of the axial flow turbine, described in [9], is shown in Fig. 10 to give the reader an idea of the performance of a bidirectional turbine in waves. Referring to Fig. 1, a sketch of the system tested leading to [9], the busbar efficiency is presented as a function of the incident wave height for the optimal operational wave frequency of 0.53 Hz. This frequency corresponds to a draft of 0.46 meter for the one-meter diameter capture chamber. The peak efficiency value is in Fig. 10 is about 36%. Because of the orientation of the capture chamber (mouth facing downward), the theoretical maximum efficiency of the system is only 50%. From the comparison of the wind tunnel tests of the two turbines, we can conclude that the radial flow turbine will outperform the axial flow turbine in operational seas. Furthermore, because of the relatively simple design of the radial flow turbine, the wave energy conversion system is more cost-effective.

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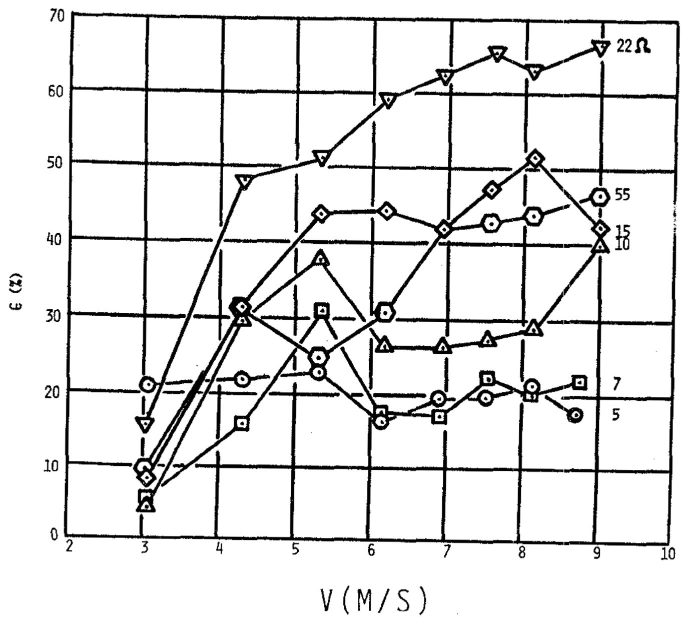

Fig. 8. Energy Conversion Efficiency for Various Electrical Loads.

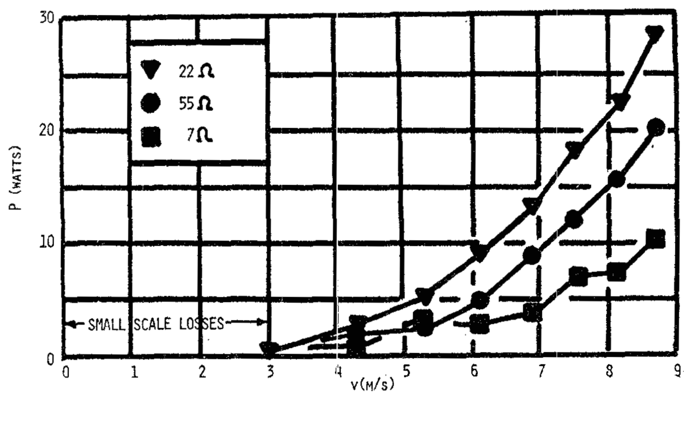

Fig. 9. Power Production for Various Three Electrical Loads.

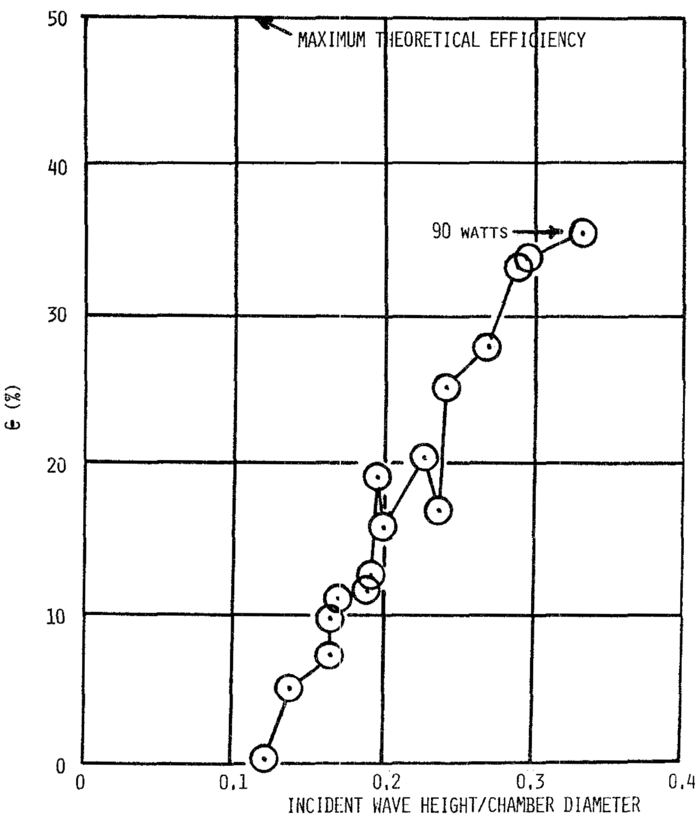

Fig. 10. Efficiency of the Counter-Rotating Axial Turbine in Wave Tank Tests.

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