where I is the compound moment of inertia. For small angles of rotation, sin θ equals θ (in radians) so that

$$\ddot{\theta} + \frac{Wl\theta}{I} = 0 \tag{13}$$

Therefore the equation of motion in roll is

$$\ddot{\theta} + \frac{K_r}{I_c}\theta = 0 \tag{14}$$

Integration of this equation between 0 and 2 yields a natural period of oscillation

$$T_r = 2\pi\sqrt{\frac{I_c}{K_r}} \tag{15}$$

and a frequency in roll of

$$f_{nr} = \frac{1}{2\pi}\sqrt{\frac{K_r}{I_c}} \tag{16}$$

As previously noted, $I_c = 0.43 \times 10^8$ ft-ton-sec² and $K_r = 85 \times 10^3 \times 14 = 1.19 \times 10^6$ ft-ton (the weight of the system times the distance between the meta center and the center of mass). Thus the natural period and frequency in roll are

$$T_r = 40.4 \text{ sec} \tag{17}$$

$$f_{nr} = 0.025 \text{ Hz} \tag{18}$$

Consider now the motion of the platform with viscous damping and forced vibration. Previous experimental work has shown that the ratio of viscous damping to critical damping is $r/r_c = 0.3$. The amplification factor (the ratio of the output angular displacement to the input angular forcing function) as a function of the ratio of the forcing frequency f to the natural roll frequency f_r is shown in Fig. 7. The forcing frequency versus angular displacement is obtained from Fig. 4. Using these two figures, one obtains the platform angular roll response versus forcing frequency shown in Fig. 8.

Examination of Fig. 8 leads one to the following conclusions:

1 The maximum roll angle of the platform is 2.8 deg for any possible forcing frequency.

2 Under sea-state-8 conditions, an extremely rare phenomenon, the maximum platform roll would be ~2.0 deg. For such waves to be possible requires a long fetch and deep water. Actually, in nearly all offshore installations, the maximum possible sea state is 6.

3 Under sea-state-6 conditions, the maximum platform roll would not exceed 0.5 deg.

Thus it may be seen that the feasibility of a nonrolling tuned platform as a nuclear-power-plant site can be shown mathematically. Experimental work on small models has demonstrated this expected result.

Reference-Design Heave Response to Sea State

The natural heave frequency F_{nh} of the reference design is computed to be 0.134 Hz. Assuming the ratio of viscous damping to critical damping to be 0.3, Fig. 6 may be applied to establish the amplification factor. The forcing function for vertical displacement is obtained from Fig. 3. Using sea-state-8 conditions, the platform heave responses in vertical displacement as a function of wave frequency are as shown in Fig. 9.

It is significant to note that under sea-state-8 conditions, maximum wave double-amplitude displacement

Fig. 7 Roll amplification versus forcing frequency/natural frequency.

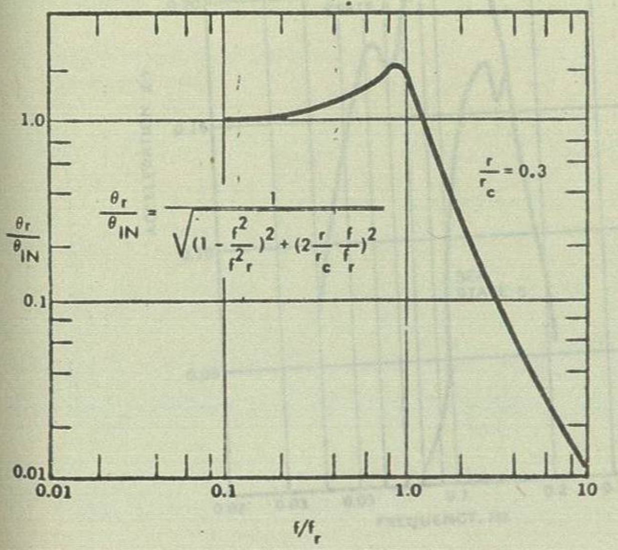
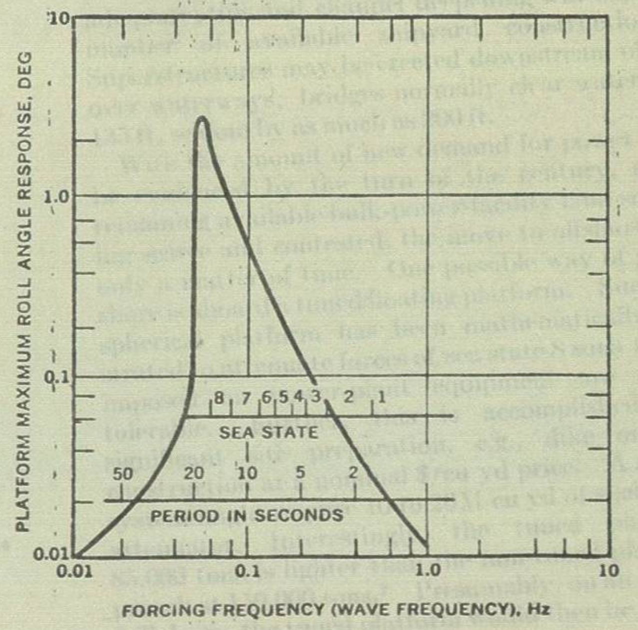


Fig. 8 Platform maximum roll angles versus forcing frequency.



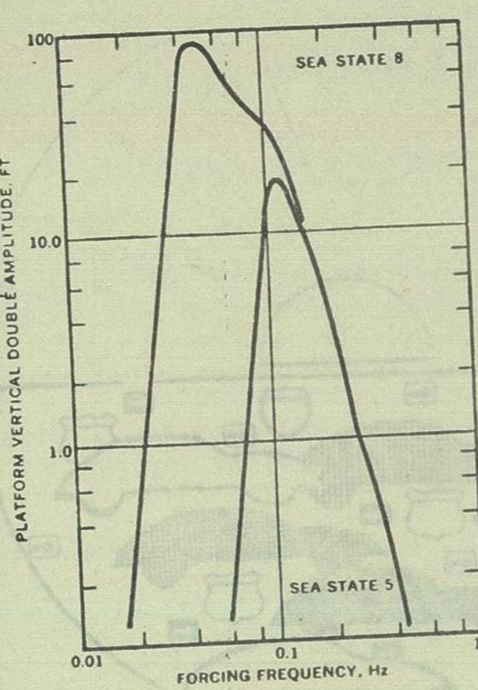


Fig. 9 Platform heave response as a function of sea state.

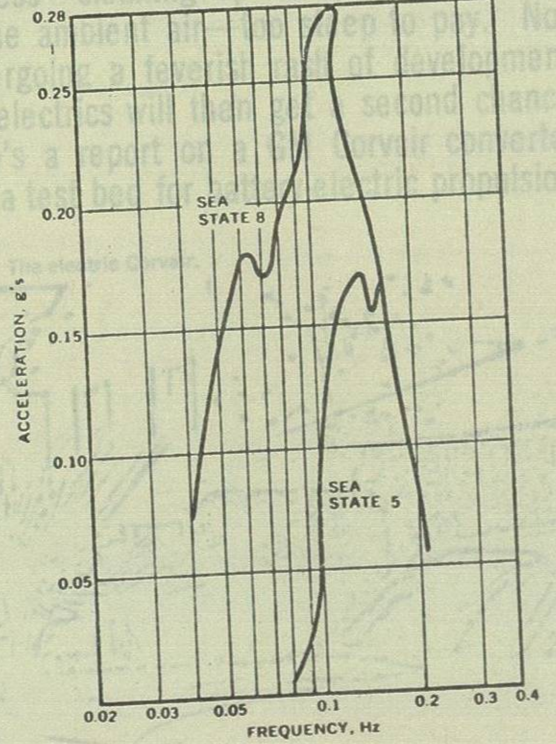


Fig. 10 Maximum acceleration experienced by platform under sea-state-8 conditions.

equals 70 ft; the double amplitude of the platform is 84 ft, or a 7-ft displacement on either side of the static water line.

The major significance of heave response is the g-loads that such motion imparts to the platform and its equipment. The acceleration experienced by the platform is

$$\ddot{a} = -a_0 w^2 \sin wt \tag{19}$$

at maximum acceleration for any forcing frequency $\sin wt = 1$. Thus

$$\ddot{x}_{\max} = -a_x w^2 \tag{20}$$

where w equals $2f$, f is the forcing frequency, and a_x is the maximum single amplitude of platform response at the forcing frequency.

Using the curve plotted in Fig. 9, for sea state 8, the maximum accelerations were derived. These data are presented in Fig. 10. It is seen that the maximum acceleration is 0.28 g's at sea state 8 and 0.17 g's at sea state 5, well within design capability of a platform and reactor/turbine-generator equipment.

Conclusions

Problems common to tuned and non-tuned platforms have not been discussed at length herein; they have been examined, however, and viable solutions are evident. These problems include:

1 Mooring: A gimbaled bridle moor which vectors mooring loads through the meta center of a scale-model tuned platform has been tested satisfactorily; vector-ing forces through the meta center retains platform tuning.

2 Collision avoidance: Lighted buoys, lighted superstructure, etc.

3 Damage control: Multiple-hull designs, multiple-compartmentation design, energy-absorbing material at and below water line, etc.

4 Construction and towing: Draft can be modified with flotation devices where channel depths are not adequate; this and channel deepening will increase the number of available shipyard construction sites. Superstructures may be erected downstream of bridges over waterways; bridges normally clear waterways by 135 ft, seldom by as much as 200 ft.

With the amount of new demand for power that will be evidenced by the turn of the century, and with remaining available bulk-power-facility land sites growing scarce and contested, the move to offshore siting is only a matter of time. One possible way of going offshore is aboard a tuned floating platform. Such a tuned spherical platform has been mathematically demonstrated to attenuate forces of sea state 8 such that loads imposed on power-plant equipment are eminently tolerable. Further, this is accomplished without significant site preparation, e.g., dike or sea-wall construction at a nominal $/cu yd price. A non-tuned system might require 10 to 20M cu yd of such sea-force attenuator. Interestingly, the tuned platform at 85,000 tons is lighter than the non-tuned-platform approach at 150,000 tons.[2] Presumably, on an equivalent $/lb basis, the tuned platform would then be less costly.

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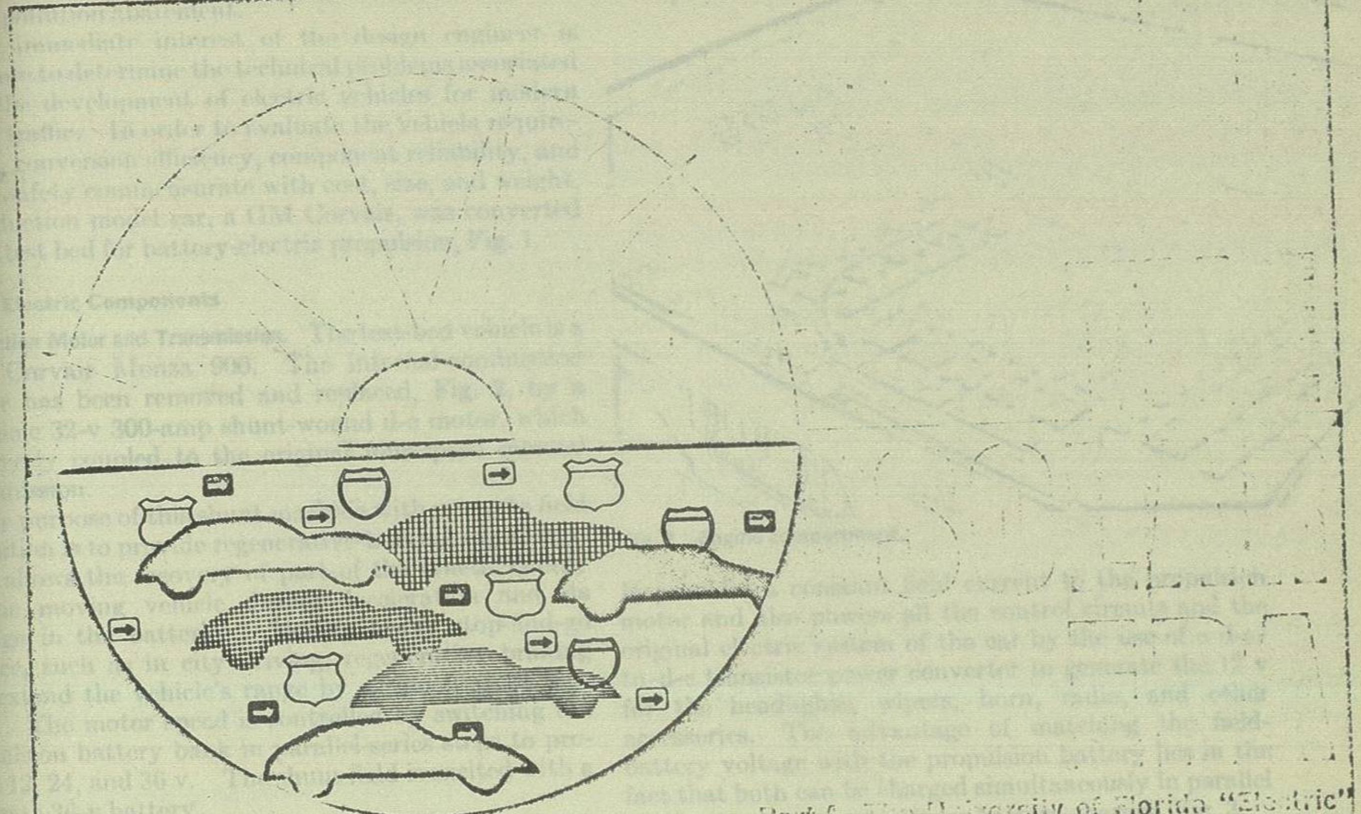
[2] *Aerospace Daily*, May 10, 1971, p. 45.



Part 4—The University of Florida "Electric"

The "electrics" are coming, or will be, if the I-C engine finds the price of its fabulous success—cleaning up its massive pollution of the ambient air—too steep to pay. Now undergoing a feverish rash of development, the electrics will then get a second chance. Here's a report on a GM Corvair converted into a test bed for battery-electric propulsion.

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RECENT developments in high-current-power transistors and silicon-controlled rectifiers combined with a new generation of high-energy electrochemical systems such as the lithium–sulfur battery have brought the

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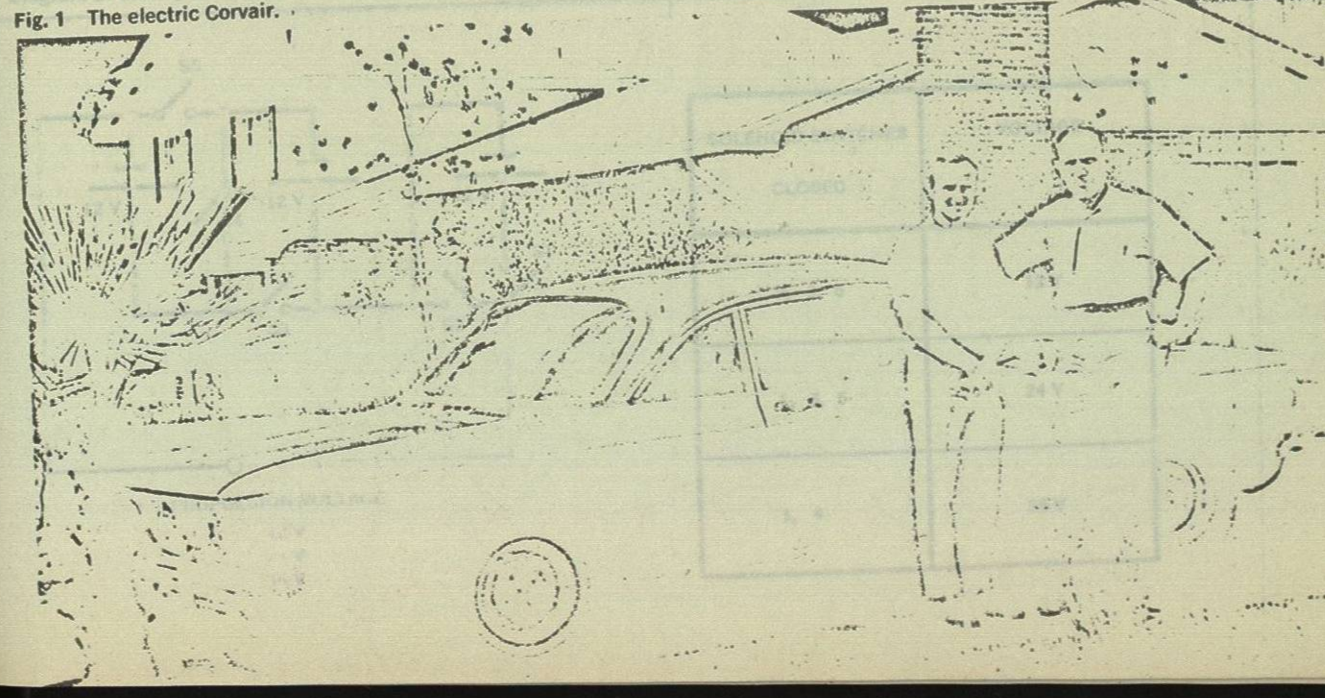
Based on a paper contributed by the ASME Solar Energy Applications Group.

Fig. 1 The electric Corvair.



"electric" into focus again as an effective means to urban pollution abatement.

The immediate interest of the design engineer is therefore to determine the technical problems associated with the development of electric vehicles for modern urban traffic. In order to evaluate the vehicle requirements, conversion efficiency, component reliability, and traffic safety commensurate with cost, size, and weight, a production model car, a GM Corvair, was converted into a test bed for battery-electric propulsion, Fig. 1.

Major Electric Components

Traction Motor and Transmission. The test-bed vehicle is a 1962 Corvair Monza 900. The internal-combustion engine has been removed and replaced, Fig. 2, by a four-pole 32-v 300-amp shunt-wound d-c motor, which is directly coupled to the original four-speed manual transmission.

The purpose of this shunt machine with separate field excitation is to provide regenerative-braking capability. This allows the recovery of part of the kinetic energy of the moving vehicle during deceleration and its storage in the batteries. During heavy stop-and-go service, such as in city driving, regenerative braking can extend the vehicle's range by as much as 25 percent. The motor speed is controlled by switching the propulsion battery bank in parallel-series steps to produce 12, 24, and 36 v. The shunt field is excited with a separate 36-v battery.

Battery System. There are two battery systems, Fig. 3, used in the electric Corvair, namely: a switchable-voltage propulsion battery and a smaller fixed-voltage field-excitation battery which also furnishes the power for the control circuits. Both batteries are of the vented nickel–cadmium type.

Propulsion Battery. The traction battery consists of three banks, 12 v each. Every bank contains 20 cells in series-parallel, thus a total of 60 cells are used. The maximum battery voltage is 36 v at 80-amp-hr capacity.

Field-excitation Battery. The field-excitation system consists of a 36-v 24-amp-hr-capacity Ni–Cd battery bank. It provides a constant field current to the propulsion motor and also powers all the control circuits and the original electric system of the car by the use of a d-c–to–d-c transistor power converter to generate the 12 v for the headlights, wipers, horn, radio, and other accessories. The advantage of matching the field-battery voltage with the propulsion battery lies in the fact that both can be charged simultaneously in parallel by the same constant-voltage battery charger, Fig. 3.

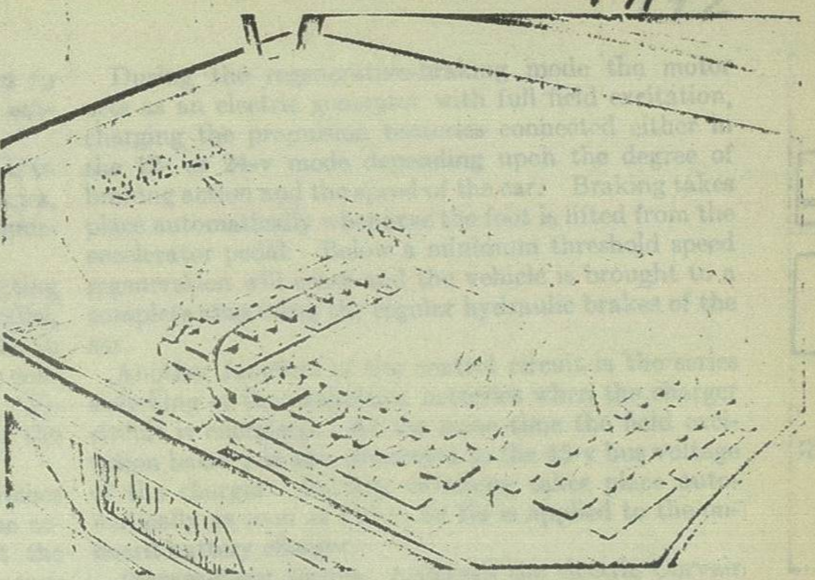
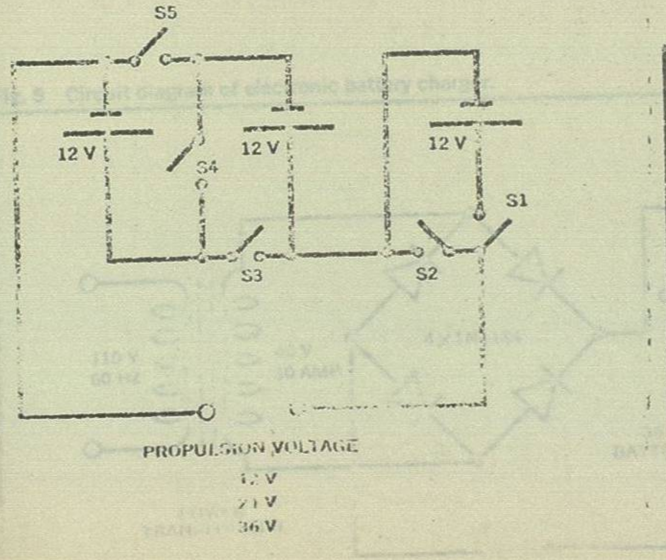


Fig. 2 Engine compartment.

Charger. Proper voltage regulation of the charger potential is important to the cell life of the battery. Too high a voltage will lead to excessive hydrogen generation, profuse gassing, spewing of electrolyte, overheating, and eventual thermal runaway, resulting ultimately in irreversible damage to the battery. Too low a voltage will cause cell imbalance requiring capacity reconditioning.

The optimum cell voltage under charge is 1.5 v at 75 F. Thus for the 36-v Corvair battery the charger voltage is maintained at 45 v electronically, Fig. 4, by means of a Zener reference diode and silicon-controlled

Fig. 3 Diagram of battery switching system.



SOLENOID SWITCHES CLOSED	VOLTAGE
2, 3, 5	12 V
1, 3, 5	24 V
1, 4	36 V