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Project Assessment & Development
John Laing Limited
Page Street
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EDINBURGH-SCOPA-LAING

LAING

5TH YEAR

WAVE ENERGY

REPORT

APPENDIX 1

MECHANICAL DESIGN PHILOSOPHY AND CALCULATIONS
<table>
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<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Profile shape</td>
<td></td>
<td>D0019</td>
</tr>
<tr>
<td>Base diameter</td>
<td>m</td>
<td>10</td>
</tr>
<tr>
<td>Hub depth</td>
<td>m</td>
<td>6.5</td>
</tr>
<tr>
<td>Water line length</td>
<td>m</td>
<td>9.9</td>
</tr>
<tr>
<td>Hydrodynamic width</td>
<td>m</td>
<td>30</td>
</tr>
<tr>
<td>Physical width</td>
<td>m</td>
<td>24</td>
</tr>
<tr>
<td>Damping coefficient</td>
<td>NmSec/rad</td>
<td>$45 \times 10^6$</td>
</tr>
<tr>
<td>Torque limit</td>
<td>Nm</td>
<td>$22.5 \times 10^6$</td>
</tr>
<tr>
<td>Power limit mean</td>
<td>MW</td>
<td>2.31</td>
</tr>
<tr>
<td>Power limit instantaneous</td>
<td>MW</td>
<td>11.53</td>
</tr>
<tr>
<td>Duck angular velocity</td>
<td>rad/Sec</td>
<td>0.89</td>
</tr>
<tr>
<td>Survival depth</td>
<td>m</td>
<td>30</td>
</tr>
<tr>
<td>Canister salvage depth</td>
<td>m</td>
<td>100</td>
</tr>
<tr>
<td>Canister diameter</td>
<td>m</td>
<td>4.8</td>
</tr>
<tr>
<td>Canister pressure</td>
<td>mmHg</td>
<td>5-10</td>
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Date: 2nd October 1979
<table>
<thead>
<tr>
<th>Specification</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of gyros</td>
<td>-</td>
<td>4</td>
</tr>
<tr>
<td>Ring cams per gyro</td>
<td>-</td>
<td>2</td>
</tr>
<tr>
<td>Cam ring diameter mean</td>
<td>m</td>
<td>4.0</td>
</tr>
<tr>
<td>Lobes per face (2 faces)</td>
<td>-</td>
<td>21</td>
</tr>
<tr>
<td>Followers per face</td>
<td>-</td>
<td>24</td>
</tr>
<tr>
<td>Width of cam face</td>
<td>mm</td>
<td>100</td>
</tr>
<tr>
<td>Mean height of over cam lobes (300 &gt; 173 mm)</td>
<td>mm</td>
<td>236.5</td>
</tr>
<tr>
<td>Weight of cam ring</td>
<td>kg</td>
<td>2375</td>
</tr>
<tr>
<td>Lobe wavelength mean at ring θ</td>
<td>mm</td>
<td>598.4</td>
</tr>
<tr>
<td>Rise of cam trough to crest</td>
<td>mm</td>
<td>63.5</td>
</tr>
<tr>
<td>Line force at a nominal pressure (18260 lb)</td>
<td>kN</td>
<td>81.23</td>
</tr>
<tr>
<td>Roller diameter</td>
<td>mm</td>
<td>185</td>
</tr>
<tr>
<td>Maximum slope tangent (25°)</td>
<td>rad</td>
<td>0.47</td>
</tr>
<tr>
<td>Minimum crest curvature radius Corrected arc</td>
<td>mm</td>
<td>384</td>
</tr>
<tr>
<td>Maximum roller acceleration</td>
<td>M/sec²</td>
<td></td>
</tr>
<tr>
<td>Line stress (4635 lb/in)</td>
<td>N/m</td>
<td>812 x 10³</td>
</tr>
<tr>
<td>Boost pressure to prevent lift</td>
<td>N/m²</td>
<td>0.345 x 10⁶</td>
</tr>
<tr>
<td>Maximum angular velocity</td>
<td>rad/Sec</td>
<td>2.67</td>
</tr>
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Date: 2nd October 1979
## SECONDARY HYDRAULICS SPECIFICATION

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of gyro drive units</td>
<td>-</td>
<td>8</td>
</tr>
<tr>
<td>Number of generator drive units</td>
<td>-</td>
<td>2</td>
</tr>
<tr>
<td>Displacement per motor revolution at maximum swash</td>
<td>m(^3)</td>
<td>2.23 x 10(^{-3})</td>
</tr>
<tr>
<td>Maximum flow into generator at 1500 rpm</td>
<td>m(^3)/Sec</td>
<td>0.1115</td>
</tr>
<tr>
<td>Maximum flow into gyros at 1500 rpm</td>
<td>m(^3)/Sec</td>
<td>0.446</td>
</tr>
<tr>
<td>Maximum total flow</td>
<td>m(^3)/Sec</td>
<td>0.558</td>
</tr>
<tr>
<td>Maximum instantaneous power (at 3000 psi) ((= 2.07 \times 10^7 \text{ N/m}^2))</td>
<td>MW</td>
<td>11.53</td>
</tr>
<tr>
<td>Maximum mean power ignoring losses</td>
<td>MW</td>
<td>2.3</td>
</tr>
<tr>
<td>Response time: zero to full flow</td>
<td>mSec</td>
<td>120</td>
</tr>
<tr>
<td>Response time: normal to double pressure</td>
<td>mSec</td>
<td>1.5</td>
</tr>
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</table>

Date: 2nd October 1979
## High Speed Gyro Bearings Specification

<table>
<thead>
<tr>
<th><strong>Description</strong></th>
<th><strong>Unit</strong></th>
<th><strong>Value</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Speed normal</strong></td>
<td>rpm</td>
<td>1500</td>
</tr>
<tr>
<td><strong>Speed max</strong></td>
<td>rpm</td>
<td>2000</td>
</tr>
<tr>
<td><strong>Maximum force</strong></td>
<td>N</td>
<td>$1.6 \times 10^6$</td>
</tr>
<tr>
<td><strong>Force angle to Polar axis</strong></td>
<td>degrees</td>
<td>18.4</td>
</tr>
<tr>
<td><strong>Axial separation of bearing couple</strong></td>
<td>m</td>
<td>4.0</td>
</tr>
<tr>
<td><strong>Diameter</strong></td>
<td>mm</td>
<td>250</td>
</tr>
<tr>
<td><strong>Length</strong></td>
<td>mm</td>
<td>500</td>
</tr>
<tr>
<td><strong>Projected pocket area effective</strong></td>
<td>m</td>
<td>0.1</td>
</tr>
<tr>
<td><strong>Shear stress</strong></td>
<td>N/m$^2$</td>
<td>$315 \times 10^6$</td>
</tr>
<tr>
<td><strong>Bending stress</strong></td>
<td>N/m$^2$</td>
<td>$2.52 \times 10^9$</td>
</tr>
<tr>
<td><strong>Deflection</strong></td>
<td>mm</td>
<td>0.276</td>
</tr>
<tr>
<td><strong>Pocket pressure nominal</strong></td>
<td>N/m$^2$</td>
<td>$138 \times 10^6$</td>
</tr>
<tr>
<td><strong>Radial land clearance</strong></td>
<td>mm</td>
<td>0.025</td>
</tr>
<tr>
<td><strong>Oil viscosity</strong></td>
<td>NSec/m$^2$</td>
<td>0.007</td>
</tr>
</tbody>
</table>
## GYRO DISC SPECIFICATION

<table>
<thead>
<tr>
<th></th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of gyros</td>
<td></td>
<td>4</td>
</tr>
<tr>
<td>Diameter at rim</td>
<td>m</td>
<td>3.05</td>
</tr>
<tr>
<td>Diameter of hole</td>
<td>m</td>
<td>1.0</td>
</tr>
<tr>
<td>Disc thickness</td>
<td>mm</td>
<td>360</td>
</tr>
<tr>
<td>Weight of disc</td>
<td>kg</td>
<td>17685</td>
</tr>
<tr>
<td>Weight of total rotating assembly extras (for shaft, 940 clamps)</td>
<td>kg</td>
<td>22025</td>
</tr>
<tr>
<td>Moment of inertia (each gyro)</td>
<td>kgm²</td>
<td>23873</td>
</tr>
<tr>
<td>Nominal working speed</td>
<td>rad/sec</td>
<td>157</td>
</tr>
<tr>
<td>Maximum stress at 1500 rpm (12 tons in²)</td>
<td>N/m²</td>
<td>185 x 10⁶</td>
</tr>
<tr>
<td>Stored energy per gyro at 1500 rpm</td>
<td>J</td>
<td>590 x 10⁶</td>
</tr>
<tr>
<td>Storage time at maximum output (0.5625 MW: from 1500 rpm to 375 rpm)</td>
<td>Sec</td>
<td>983</td>
</tr>
<tr>
<td>Maximum bearing force</td>
<td>N</td>
<td>1.6 x 10⁶</td>
</tr>
<tr>
<td>Bearing diameter (nominal)</td>
<td>mm</td>
<td>250</td>
</tr>
<tr>
<td>Bearing length (nominal)</td>
<td>mm</td>
<td>500</td>
</tr>
<tr>
<td>Projected bearing area</td>
<td>m²</td>
<td>0.125</td>
</tr>
<tr>
<td>Nominal bearing clearance radial</td>
<td>(.001&quot;)</td>
<td>0.025</td>
</tr>
<tr>
<td>Precession gear ratio</td>
<td></td>
<td>3:1</td>
</tr>
<tr>
<td>Iw (whole duck)</td>
<td>kgm²/sec</td>
<td>15 x 10⁶</td>
</tr>
<tr>
<td>Energy per gyro at maximum storage speed (2000 rpm)</td>
<td>J</td>
<td>1.05 x 10⁷</td>
</tr>
<tr>
<td>Prestresses for 2000 rpm (32 tsi compressive: 16.7 tsi tensile)</td>
<td>N/m²</td>
<td>4.94 x 10⁷</td>
</tr>
<tr>
<td><strong>PRIMARY HYDRAULIC SPECIFICATION</strong></td>
<td>Unit</td>
<td>Value</td>
</tr>
<tr>
<td>------------------------------------</td>
<td>------</td>
<td>-------</td>
</tr>
<tr>
<td>Number of lobe stations per cam ring (2 lobes per station)</td>
<td>-</td>
<td>21</td>
</tr>
<tr>
<td>Number of pumping stations per cam (2 cylinders per station)</td>
<td>-</td>
<td>24</td>
</tr>
<tr>
<td>Number of ring cam pumps per gyro (4 gyros per duck)</td>
<td>-</td>
<td>2</td>
</tr>
<tr>
<td>Primary cylinders per duck (2 x 24 x 2 x 4)</td>
<td>-</td>
<td>384</td>
</tr>
<tr>
<td>Bore</td>
<td>mm</td>
<td>50</td>
</tr>
<tr>
<td>Stroke</td>
<td>mm</td>
<td>127</td>
</tr>
<tr>
<td>Nominal radial clearance (0.001&quot;)</td>
<td>mm</td>
<td>0.025</td>
</tr>
<tr>
<td>Volume per cylinder stroke</td>
<td>m³</td>
<td>249 x 10⁻³</td>
</tr>
<tr>
<td>Volume per cam revolution per each ring cam pump (2 x 249 x 10⁻³ x 24 x 21)</td>
<td>m³</td>
<td>0.251</td>
</tr>
<tr>
<td>Work rate per cylinder per MW</td>
<td>Hz</td>
<td>194</td>
</tr>
<tr>
<td>Working pressure in HP Main (3000 psi)</td>
<td>N/m²</td>
<td>2208 x 10⁶</td>
</tr>
<tr>
<td>Boost pressure in LP Main (50 psi)</td>
<td>N/m²</td>
<td>345 x 10⁶</td>
</tr>
<tr>
<td>Power strokes per cylinder in 25 years</td>
<td>-</td>
<td>2.39 x 10⁸</td>
</tr>
<tr>
<td>Oil viscosity (7 centistokes)</td>
<td>NSec/m²</td>
<td>0.007</td>
</tr>
</tbody>
</table>
Ring Cam Pump

Leakage and Shear Power Loss

Calculations:

1. Piston/Sleeve
   Page 2
2. Big end, little end of Con Rod
   Page 4
3. Lever Pinot Bearing
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4. Cam Follower Roller
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5. Gyro Cage Rocker Bearings
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Add'l Loss and Misc. Calculations

1. Cam Follower Roller rolling losses
   Page 16
2. Minimum Ring Cam Pump Boost Pressure
   Page 17

Conclusions

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Ring Cam Pump

Piston/Sleeve Power Losses

Characteristics
Dia: 50 mm (1.969")
Length of Skirt = 60 mm (2.362")
Radial clearance: .001"

Leakage Losses (100% capacity)

\[ R_h = \frac{12 \times 2.36 \times 3 \times 10^{-6}}{(\pi \times 1.969) \times 0.001^3} = 13735 \]

\[ Q = \frac{\Delta P}{R_h} = \frac{3050}{13735} = 0.22 \text{ cips/piston} \]

Leakage Power/gyro (inc. cycling)

\[ Q = 0.22 \times 9.6 \times \frac{1}{2} = 10.56 \text{ cips} \]

\[ P_L = \frac{10.56 \times 3050 \times 746}{850 \times 12} = 3.64 \text{ kW/gyro} \]

For radial clearance:
.007"

\[ P_L = 1.25 \text{ kW/gyro} \]

.005"

\[ P_L = .46 \text{ kW/gyro} \]
**Shear Losses** (Independent of Output)

\[ A = 2.36^{\text{\#}} \times 1.97 \times \pi \]
\[ = 14.61 \text{ in}^2 \]

if the cycle is approximated by a sinusoidal then \( V_{\text{avg}} = \frac{21 \text{ lb x 170}^\circ}{360 \times 10 \text{ ft/min}} \)

\[ V_{\text{rms}} = \frac{\pi}{\sqrt{2}} V_{\text{avg}} = \frac{\pi}{\sqrt{2}} \times 19.83 = 44.1 \text{ in/sec} \]

\[ h = 0.001 \]
\[ P_S = \frac{A x V_{\text{rms}} x V}{6600 x h} \]
\[ P_S = \frac{14.61 \times 4.41^{\text{\#}} \times 3 \times 10^6}{6600 \times 0.001} \times 96 \times 0.746 = 0.92 \text{ KW/gro} \]

\[ h = 0.007 \]
\[ P_S = 1.32 \text{ KW/gro} \]

\[ h = 0.005 \]
\[ P_S = 1.84 \text{ KW/gro} \]

**Piston/Sleeve Power Loss/gro**

\[ P = P_L \times \% \text{ operative cyl} + P_S \]

\[ 127\% \text{ - operative capacity in ring cam, at full power} \]

\[ 79\% \text{ of cylinders are working} \]

for \( h = 0.001" \text{ radial clearance} \)

\[ 100\% \text{ - full power loss/gro} \]

\[ P = 3.64 x \% \text{ op} + 0.92 \]
\[ P = 3.64 \times 79 + 0.92 = 3.80 \text{ KW} \]

for \( h = 0.007 \)
\[ P = 1.25 x \% \text{ op} + 1.32 \]
\[ P = 1.25 \times 79 + 1.32 = 2.31 \text{ KW} \]

for \( h = 0.005 \)
\[ P = 0.42 x \% \text{ op} + 1.84 \]
\[ P = 0.42 \times 79 + 1.84 = 2.71 \text{ KW} \]
Hydrostatic Big End 8rg

Loading  \[ F = 9280 \text{ lb} \]

**Characteristics:**

Ball Dia. = 2.375"

radial clearance = .0005"

loaded def. = .0002"

Angular excursion = 8° from top to bottom of stroke.

**Bottom Pocket Area:**

\[
A_b = \frac{\pi}{4} \left( 2.375 \times \sin 2\times 25^\circ \right)^2 = 0.75 \text{ in}^2.
\]

Projected area of peripheral pocket

\[
A_r = \frac{\pi}{4} \times (2.375^2 \left( \sin 47.5^\circ \right)^2) = 2.88 \text{ in}^2.
\]

Load supported by bottom pocket

\[
L = PA_b = 3050 \times 0.75 = 2288 \text{ lb}
\]
Peripheral Pocket Pressure

\[ P_p = \frac{9270 - 2288}{2.88} = 2424 \text{ psi} \]

Leakage between pockets.

\[ R_h = \frac{12 \times 32 \times 3 \times 10^{-6}}{\pi \times 2.18 \times \frac{0.003^3}{3}} = 86,850 \]

\[ Q = \frac{\Delta P}{R_h} = \frac{3050 - 2500}{86,850} = 0.006 \text{ cips.} \]

Leakage out of Peripheral Pocket

\[ R_h = \frac{12 \times 32 \times 3 \times 10^{-6}}{\pi \times 2.18 \times \frac{0.003^3}{3}} = 63,930 \]

\[ Q = \frac{2500}{63,930} = 0.039 \text{ cips.} \]

Flow Through Impedence into Per Pocket

\[ Q = 0.039 - 0.006 = 0.033 \text{ cips} \]

\[ R_{th} = \frac{\Delta P}{Q} = \frac{3050 - 2500}{0.033} = 16,620 \]

Leakage Power Loss / gyro

\[ P_L = \frac{0.033 \times 3050 \times 96 \times 746}{550 \times 12} \times \frac{1}{2} \]

\[ P = 0.54 \text{ kw/gyro.} \]
Shear Power Losses

If the connecting rod is assumed to move with simple harmonic motion

period = 5 sec    gyro rotation = 170°
can lobes = 21

\[ F = \frac{21 \times 170}{360} \times \frac{1}{5} = 1.78 \text{ cycles/sec} \]

avg. surface speed of ball = \[ V = \frac{2 \times 2.8 \pi \times 1.78}{2 \times 1.80} \]

\[ V = 0.66 \text{ in/sec} \]

\[ V_{rms} = \frac{\pi}{\sqrt{2}} V = \frac{\pi}{\sqrt{2}} \times 0.66 = 1.47 \text{ in/sec} \]

\[ P_s = \frac{A V_{rms} R}{6600 \times h} \]

\[ A = 0.32 \times 2.15 \pi + 0.20 \times 0.98 \pi \]

\[ A = 2.78 \text{ in}^2 \]

\[ P_s = \frac{2.78 \times 1.47^2 \times 3 \times 10^{-6} \times 96 \times 2.746}{6600 \times 0.0005} \]

\[ P_s = 0.4 \text{ watt/gyro, negligible} \]

Small End.

Leakage losses are negligible as it is a film berg.

Shear losses are negligible as the movement is even smaller than the big end (2°) and the diameter is less as well.
Hydrostatic Lever Borg.

Borg: Dimensions
Dia: 3" Width: 6.4"

Joint RMS Velocity
roller excursion 2.5"
angular excursion = \( \frac{2.5}{7.5} = 0.2 \text{ rad} \)
cam runs angular vel. = 1.89 rad/sec
cam dia. = 4 m (157.5") lobe wave length = 23.6"
time/lobe = \( \frac{23.6 \times \pi}{1.89 \times 157.5} = 0.16 \text{ sec/lobe} \)
avg vel. during one cycle \( \frac{2 \times 2}{0.16} = 2.52 \text{ rad/sec} \)
= \( \frac{0.40 \text{ rev}}{5 \text{ sec}} \)
due to low speeds: select wide lands
try 6-5" c = 0.005"

Projected main pad area
\[ A = \left( (2.58 - 0.40) \times 5.9 \right) = 11.70 \text{ in}^2 \]

Projected Tang pad area
\[ A = (2.03 - 0.4) \times 5.9 = 9.60 \text{ in}^2 \]
**Pressures**

1. Main Brg

\[ P = \frac{9272}{11.70} = 782 \text{ psi} \]

2. Tang Brg (differential)

\[ P = \frac{8650}{9.60} = 901 \text{ psi} \]

Use 800 psi as pad pressure

**Leakage Impedence**

1. Main Brg

\[ R_h = \frac{12 \times 5 \times 3 \times 10^{-6}}{(0.005)^3} = 9120 \text{ cm}^3/\text{sec} \]

2. Tang Brg

\[ R_h = \frac{12 \times 5 \times 3 \times 10^{-6}}{(0.005)^3} = 9800 \text{ cm}^3/\text{sec} \]

\[ R_h = \frac{12 \times 10^{-6}}{(0.003)^3} = 9940 \text{ cm}^3/\text{sec} \]

\[ R_h = \frac{12 \times 10^{-6}}{(0.0017)^3} = 244 \text{ cm}^3/\text{sec} \]

**Leakage**

1. Main Brg.

\[ Q = \frac{800}{9120} = 0.088 \text{ cips} \]

2. Tang Brg.

\[ Q = \frac{900}{9600} = 0.092 \text{ cips} \]

**Threaded Impedence**

1. Main Brg.

\[ R_{th} = \frac{\Delta P}{Q} = \frac{3050-800}{0.088} \]

\[ R_{th} = 25570 \text{ cm}^3/\text{sec} \]

2. Tang Brg.

\[ R_{th} = \frac{3050-400}{0.042} \]

\[ R_{th} = 63095 \text{ cm}^3/\text{sec} \]
Leakages

1. Main long. (cycle including 1/4 time at boost/press)

\[ Q = \frac{1}{2} \left( \frac{3050 + 50}{25570 + 9120} \right) = 0.95 \text{ cips.} \]

2. Tang long (cycle 1/4 time under side load, 1/4 time at press, 1/4 time at boost)

\[ Q = \left( \frac{0.25}{44440 + 63075} + \frac{0.25}{244 + 63075} + \frac{0.50}{9600 + 63075} \right) \times \frac{1}{2} \times 2 \times 3050 \]

\[ Q = 0.040 \text{ cips.} \]

\[ Q_f = 0.040 + 0.045 = 0.085 \text{ cips} \]

**Power Loss in Leakage**

\[ P_L = 0.085 \times 3050 \times 96 \times 0.746 \times \frac{350}{12} = 2.8 \text{ kW/gyro} \]

**Shear Losses/gyro**

\[ b = \left( \frac{\pi \times 3 \times 1}{3} \right) + (6 \times 5 \times 5.4) = 2.72 \]

\[ P_5 = \frac{275 \times \pi^3 \times 3^3 \times 0.40^2 \times 2.72 \times 96 \times 0.746 \times 3 \times 10^{-6}}{300 \times 6600 \times 0.0005}. \]

\[ P_5 = 0.018 \text{ kW} \]

**Power Loss for Pivot/gyro**

\[ P = P_5 + P_L = 2.72 + 0.02 = 2.74 \text{ kW/gyro} \]
Cam Follower Roller Bearing

Rotational Speed

Cam Ring Development ratio
(true rolling length) = 1:0.5

Roller dia = 185 mm  Cam Dia = 4 M.

Roller velocity ratio = \( \frac{1.05 \times 4000}{185} = 227 \)

Cam ring RMS Velocity = 1.89 rad/sec
Roller RMS Velocity = \( \frac{1.89 \times 227}{\text{sec}} = 4286 \text{ rad/sec} \)
\[ = 6.82 \text{ rev/sec} \]

Loading

\( \phi = 25^\circ \) at max loading condition

\[ F_P = 9272 \text{ lb} \]
\[ F_R = \frac{15}{7.5} F_P = 2 \times 9272 \]
\[ F_R = 18544 \text{ lb} \]

\[ F_m = \frac{F_R}{\cos \phi} = \frac{18544}{\cos 25^\circ} = 20460 \text{ lb} \]

\[ F_t = F_m \sin \phi = 20460 \sin 25^\circ = 8650 \text{ lb max} \]
**Brig. Characteristics**

- **Dia.:** 4.1 in.
- **Width:** 4.33 in.
- **Radial clearance:** .0005".
- **Land Width:** .25 in.
- **Equatorial land width:** .188 in.

**Main Brig.**

\[
\text{Area} = 3.93 \times (4.33 - .25) = 14.00 \text{ in}^2
\]

\[
\text{Pressure} = \frac{L}{A} = \frac{18544}{14.0} = 1320 \text{ psi}
\]

Use 1300 psi as pad pressure.

**Tang. Brig.** (Note this is differential type)

\[
\text{Area} = 1.72 \times 4.1 = 7.0 \text{ in}^2
\]

1. **Differential Pressure (Max) = \frac{8650}{7.0} = 1235 \text{ psi}**

We will use a much lower pad pressure than this as this is the maximum differential req'd and will be achieved when one side of the brig. closes up.

\[\therefore \text{use 400 psi as pad pressure}\]
Leakage Impedences

Main:
\[ R_h = \frac{12 \times h \cdot \sqrt{V}}{I \times h^3} = \frac{12 \times 2.5 \times 3 \times 10^{-6}}{(2 \times 4.3 + 2 \times 0.4) \times (0.005)^3} \]

\[ R_h = 4.340 \]

Tangential Brg.
\[ h = 0.0005 \] \[ R_h = \frac{12 \times 2.5 \times 3 \times 10^{-6}}{(2 \times 2.1 + 2 \times 0.4) \times (0.005)^3} = 5900 \]

\[ h = 0.003 \] \[ R_h = \frac{12 \times 2.5 \times 3 \times 10^{-6}}{12 \times 20 \times (0.003)^3} = 7320 \]

\[ h = 0.007 \] \[ R_h = \frac{1.8 \times 10^{-6}}{12 \times 20 \times 0.0007^3} = 2150 \]

Leakages

1. Main Brg.
\[ Q = \frac{P}{R_h} \quad Q = \frac{1200}{4340} = 0.28 \text{ cips} \]

\[ Q = \frac{400}{5900} = 0.068 \text{ cips} \]

Threaded Impedences

1. Main
\[ R_{th} = \frac{\Delta P}{Q} = \frac{3050 - 1320}{28} = 6250 \]

2. Tang.
\[ R_{th} = \frac{3050 - 900}{0.068} = 38970 \]
**ProJ No.**  
**Sheet No.** 12  
**Prep. by**  
**Date**  

---

**INGECO LAING**  
**CALCULATIONS**  

**Brg. Leaktages (Actual, including Cycling)**  

1. **Main**  
   \[ Q = \frac{1}{2} \times \frac{3050}{6250 + 4340} = 0.144 \text{ cips} \]

2. **Tangential (including \( \frac{1}{4} \) cycle at \( 0.002'' \) defl.)**  
   \[ Q = 3050 \left[ \frac{0.25}{27300 + 39000} + \frac{0.25}{2150 + 39000} \right] \]
   \[ Q = 0.064 \text{ cips.} \]

**Leakage Power Loss / gyro (all cysls operating)**  

\[ P_L = \left( 0.144 + 0.064 \right) \times 3050 \times 96 \times 0.746 \text{ kw} \]
\[ 550 \times 12 \]

\[ P_L = 6.88 \text{ kw} \]  
*(note: this will drop in proportion 5.44 kw at full output with load)*

**Shear Power Loss / gyro**  

\[ P_s = \frac{\pi^3 D^3 b H^2 V}{6600 \times H} \]  
*equatorial load*

\[ b = (\pi \times 4.1) \times 5 + (\pi \times 4.1) \times 1.13 + 6 \times 25 \times 3.8 = 113'' \]
\[ \pi \times 4.1 \]

\[ P_s = \frac{\pi^3 \times 4.1^3 \times 6.82^2 \times 1.13 \times 3 \times 10^{-6} \times 96 \times 0.746 \times 125}{6600 \times 0.005} \]
\[ = 4.9 \text{ kw/gyro.} \]
Gyro Precession Rocking Brs.

Velocity

\[ \omega = \frac{2\pi \times 170^\circ}{360^\circ} \times \frac{1}{5 \text{ sec}} = 0.59 \text{ rad/sec} \]

\[ \omega_{rms} = 0.59 \times \frac{\pi}{\sqrt{2}} = 1.32 \text{ rad/sec} \quad \text{(for SHM)} \]

\[ n = \frac{\omega_{rms}}{2\pi} = 0.21 \text{ rev/sec} \]

Loading

\[ F_R = 160 \text{ T} \times \frac{13}{14.6} \]

\[ F_c = 142 \text{ T} \quad \text{(at Torque limit)} \]

Brg. Characteristics

Spherical Journal: 21" Dia. x 15" Wide

Projected Area = 2857.7 in² (190° prime pads)

Projected Land Area = 42 sq in.

Cross Land Area = 13.23 \times 2 = 6.0 in²

Big Area (Effective)

\[ A = \sin 20^\circ \left( 286 - \frac{4.2}{2} \right) - \frac{1}{2} (6.0) \]

\[ A = 246 \text{ in}^2 \]
Differential Pressure

\[ \Delta P = \frac{142 \times 2240}{246} = 1293 \text{ psi} \]

Use pad pressure of 1500 psi.

Leakage Impedence

\[ R_h = \frac{12 \times b \sqrt{v}}{\pi h^3} = \frac{12 \times 1.33 \times 3 \times 10^{-6}}{15.94 \pi \times 2 \times .001^3} = 478 \]

Leakage

\[ Q = \frac{1500}{478} = 3.14 \text{ cips.} \]

Threaded Impedence

\[ R_{th} = \frac{\Delta P}{Q} = \frac{3050 - 1500}{3.14} = 999 \]

Leakage Loss

\[ P_L = \frac{3.14 \times 3050 \times 2 \times .746}{550 \times 12} = 2.16 \text{ KW/gyro} \]

Shear Loss

\[ P_s = \frac{\pi^3 D^3 L^2 b \mu}{6600 \lambda} \]

\[ 6 = 1.33 + \frac{.5 \times 132 \times 2}{15.94 \pi} \]

\[ P_s = 1.59 \]

\[ P_s = \frac{\pi^3 1594^3 \times .21 \times 1.59 \times 3 \times 10^{-6} \times 2 \times .746}{6600 \times .001} \]

\[ P_s = .006 \text{ KW/gyro} \]
Follower Roller Rolling Resistance

\[ P = f W \frac{V_m}{n} \]

- \( f = 0.011 \) for clay soil
- \( W = 20,000 \text{ lb} \) (1/2 cycle unloaded)
- \( V_m = \frac{170}{360} \times \frac{1}{5} = \frac{4}{5} \times 3.89 = 3.08 \text{ ft/sec} \)

\[ P = 0.011 \times 20,000 \times 3.08 \times \frac{1}{550} = 0.746 \times 96 \]

\[ P = 4.4 \text{ kW/gyro} \]
Determination of Ring Cam

Boost Pressure

Boost pressure is determined by the resistance req'd to prevent the unloaded roller from leaving the cam surface by virtue of the acceleration and inertia of the swing arm and its attachments.

Moments of Inertia. (swing arm is alu)

\[ I = \frac{W}{9} \frac{a^2 + b^2}{12} \]
\[ W = 75 \times 9 \times 5.25 \times 12 \]
\[ W = 2816 \]

\[ I_e = \frac{28}{362} \frac{6^2 + 7.5^2}{12} = 6.77 \text{ in}^2 \text{sec}^2 \]

\[ W = 9 \times 4.5 \times 5.25 \times 12 = 25516 \]

\[ I_e = \frac{25.5}{362} \frac{4.5^2 - 9^2}{12} = 6.7 \text{ in}^2 \text{sec}^2 \]
Roller + Brgy

Roller is in Stl.

\[ I = \frac{W}{g} k^2 \]

\[ W = 2.86 \times 4.25 \times \frac{\pi}{4} \left( 7.25^2 - 2.5^2 \right) \]

\[ I = 34 \text{ lb-in-sec}^2 \]

Connecting Rod (Stl)

Rotational inertia negligible

\[ W = 2.86 \times \left( 6 \times 0.37^2 \times \frac{\pi}{4} + \frac{4}{3} \times 2 \times 2375 \right) \]

\[ W = 3.87 \text{ lb} \]

Assembly

\[ I_{\text{Total}} = 6.77 + \frac{2.8}{32.2} \times 4^2 + 6.7 + \frac{25}{32.2} \times 11.2^2 + \frac{34}{32.2} \times 7.5^2 \]

\[ + 0.44 + \frac{6.7}{32.2} \times 7.5^2 + \frac{3.87}{32.2} \times 15^2 \]

\[ I_{\text{Total}} = 225 \text{ in-lb-sec}^2 \]
Acceleration can be approximated by sine curve

\[ y = 1.25 \sin \left( \frac{x}{3.75} \right) \]
\[ \dot{y} = -1.25 \cdot 3.75 \cos \left( \frac{x}{3.75} \right) \]
\[ \ddot{y} = 1.25 \cdot 3.75^2 \sin \left( \frac{x}{3.75} \right) \]
\[ \dddot{y} = \text{max at } \sin \left( \frac{x}{3.75} \right) = 1 \]
\[ \dddot{y} = -17.58 \text{ in/sec}^2 \]

\[ \alpha = \frac{y}{r} = \frac{17.52}{7.5} = 2.34 \text{ rad/sec}^2 \]

\[ T = I \alpha = 225 \text{ in}-\text{lb} \cdot \text{sec}^2 \cdot 2.34 = 526.50 \text{ in-lb} \]

**Piston Force Req'd to oppose inertia force**

\[ F = \frac{I}{r} = \frac{526.5}{15} = 35 \text{ lb} \]

Piston Area = 3.04 in\(^2\)

**Burst Pressure (Minimum)**

\[ P = \frac{35}{3.04} = 11.5 \text{ psi} \]
Conclusions

The total ring cam pump losses can be divided into two groups
1. Constant and 2. those varying with load.

<table>
<thead>
<tr>
<th>Item</th>
<th>Constant Losses</th>
<th>Load Dependent Losses</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston/Sleeve</td>
<td>3.7 KW/Duck</td>
<td>14.6 KW/Duck</td>
</tr>
<tr>
<td>Con Rod</td>
<td>0.0</td>
<td>2.2 KW/Duck</td>
</tr>
<tr>
<td>Lever Pivot</td>
<td>0.1</td>
<td>11.2 KW/Duck</td>
</tr>
<tr>
<td>Roller</td>
<td>19.6</td>
<td>27.5+223 kw.</td>
</tr>
<tr>
<td>Gyro Rocker Brgs.</td>
<td>0.0</td>
<td>8.64 kw.</td>
</tr>
<tr>
<td>Total</td>
<td>23.3</td>
<td>81.74 KW/Duck</td>
</tr>
</tbody>
</table>

\[ P_{\text{100}} = 23.3 + 0.73 \times 81.74 \]
\[ P_{\text{100}} = 83.0 \text{ KW/Duck} \]

\[ P_{\text{83}} = 72.8 \]
\[ P_{\text{67}} = 63.3 \]
\[ P_{\text{50}} = 53.1 \]
\[ P_{\text{33}} = 43.0 \]
\[ P_{\text{25}} = 38.2 \]
King Cam Pump

Power Conversion Characteristics

Cylinder Displacement

\[ V_{cy} = 5'' \times 1.969^2 \times \frac{\pi}{4} \]
\[ = 15.22 \text{ in}^3 \]

at 1.98 cycles/sec

Disp = 1.98 \times 5.22 = 30.14 cips.

Compressibility losses

(Volumetric, not Power)

\[ \eta = 1 - \frac{P \times V^4}{V_{cy}} x C \]
\[ \text{where } C = \frac{\text{ratio of vol. change}}{\text{pressure}} \]
\[ \text{(assuming a linear compressibility)} \]

\[ \eta = 1 - \frac{3150 \times (152-32)}{15.2} \times \frac{.01}{2200} \]
\[ \text{cyl. head vol: 32 in}^3 \]

\[ \eta = .956 \]

Gross Cylinder Volumetric Output

\[ Q_{cy} = 30.14 \times .956 = 28.8 \text{ cips} \]

Gross Power output (disregarding leakage and shear losses)

\[ P = \frac{28.8 \times 96 \times 4 \times 3150}{550 \times 12} \times .746 \]

\[ P = 3938 \text{ kW} \]
Net Full Power Output

\[ P_{nf} = P - (P_f + P_l) = 3738 - (2174 + 233) \]

\[ P_{nf} = 3833 \text{ kW.} \]

Net Flow

\[ Q_{net} = 23.8 \times 96.4 - \frac{64.1 \times 550 \times 12}{0.746 \times 3150} \]

\[ Q_{net} = 19,880 \text{ cips.} \]

Instantaneous Flow Rate

Max. Ring Cam Velocity = 2.67 rad/sec

Cycles/sec = \( \frac{2.67 \times 1 }{2 \pi} \times 21 \text{ cycles/rev.} = 8.92 \)

\[ Q_{inst} = 1522 \times 8.92 \times 0.956 \]

\[ Q_{inst} = 129.8 \text{ cips.} \]

\[ Q_{inst\text{ Total}} = 129.8 \times 96.4 = 180 \]

\[ Q = 49,680 \text{ cips} \]

Input Flow Capacity of High Speed Motors

\[ = 3400 \text{ cips} + 28.5 \text{ (slip loss)} = 3430 \]

Total Input Capacity

\[ = 10 \times 3430 = 34,300 \text{ cips.} \]
Redundancy of Ring Cam

*Instantaneous* (based on flow)

\[ \% R = 1 - \frac{39,300}{49,680} = 31\% \text{ of cyls inoperative} \]

*Average Redundancy* (at full power)

\[ \% R = 1 - \frac{2802}{3833} = 27\% \text{ cyls inoperative} \]
Efficiency of the Mechanical-Hydraulic-Electrical Power Conversion System Vs. Load
**Flywheel Analysis**

Introduction: The force system on the flywheel consists of three separate elements:

1. Centrifugal force due to rotation
2. Prestressing forces to counteract the centrifugal forces
3. Gyro forces from the rocking and precessing motions

It is intended to handle each of these systems separately and then combine them by the method of superposition to determine the maximum stresses.
Flywheel Characteristics

\[ D_0 = 10 \text{ ft} \quad D_1 = \quad \phi = 10^\circ \]

\[ D_1 = 39.4'' \quad D_0 = 120'' \]

1. Centrifugal Force System

\[ \min N = 2000 \text{rpm} \quad \frac{N}{\text{rev sec}} = 333 \quad \omega = 209 \text{ rad sec} \]

Operating \[ N = 1500 \text{ rpm} \quad \frac{N}{\text{rev sec}} = 25 \quad \omega = 157 \text{ rad sec} \]

As the centre of the disc is restrained, the formulae for thin discs are applicable.

\[ \sigma_r = \frac{\left(3 + \nu\right)}{8} \frac{W}{\rho} \omega^2 \left(r_o^2 - r^2\right) \]

\[ \sigma_\theta = \frac{W}{8} \frac{\omega^2}{\rho} \left[(3 + \nu) r_o^2 - (1 + 3\nu) r^2\right] \]

\[ W = \frac{\text{wt}}{\text{unit vol}} = \frac{.2816}{\text{in}^3} \quad \nu = .3 \]

\[ \sigma_r = \left(\frac{3 + 3}{8}\right) \left(\frac{.28}{322.12}\right) \left(\frac{209}{157}\right) \left(60^2 - r^2\right) = \left\{\begin{array}{c} 13.1 \ (3600 - r^2) \\ 7.37 \ (7.37) \end{array}\right\} \]

\[ \sigma_\theta = \left(\frac{.28}{8}\right) \left(\frac{\left(\frac{109}{72}\right)^2}{322.12}\right) \left[(3 + 3) 60^2 - (1 + 7) r^2\right] = \left\{\begin{array}{c} 4.0 \ (11,880 - 19 r^2) \\ 2.2 \ (2.2) \end{array}\right\} \]
<table>
<thead>
<tr>
<th>r (in)</th>
<th>( \sigma_r ) (ksi)</th>
<th>( \sigma_0 ) (ksi)</th>
<th>157 rad/sec</th>
<th>209 rad/sec</th>
</tr>
</thead>
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<tr>
<td>27.6</td>
<td>10.3</td>
<td>9.3</td>
<td>17.6</td>
<td>16.6</td>
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<td>18.5</td>
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<tr>
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<td>0</td>
<td>9.0</td>
<td>0</td>
</tr>
</tbody>
</table>

**Centrifugal Stress vs. radius** 276"

**Stress in TSI.**

**512"**

**60"**
2. Prestressing of the Flywheel Discs.

A method of analysis of the stresses induced in disc flywheels was developed by R. Clerk but has been lost. A replacement is not yet available but is at present being redeveloped.

This prestressing will give the disc centre a compressive stress approximately three times higher than the accompanying tensile stress at the rim. As the stress is in different directions at either boundary it will be noted that a nodal point of zero circumferential stress exists somewhere between the two, thus the centrifugally induced stresses will be unopposed here. The stress at this point will therefore serve as a target stress for the superposed maximum (of centrifugal + prestress stresses). This is borne out by R. Clerk's paper for the 1977 Symposium on Flywheels.
3. *Swashloads* (from gyroscopic forces)

The gyroscopic precession and dock foring loads on the flywheel impart a torque on the flywheel perpendicular to its axis of rotation. The leading that it contributes as a whole to the shaft is known, and it is proposed to develop the loading on the flywheel from these forces.

The analysis of asymmetrical loading on a structure such as this flywheel is extremely complex and must, almost certainly be done by numerical methods (i.e. finite element). In the small time given to this analysis this is not possible, so that a very approximate analysis is used. This is intended solely to show that the wobble forces in the flywheel are very low, even if they are simulated by a much weaker type of structure which can be analyzed. If these forces show themselves to have any great magnitude the results will not be meaningful, except to show that a better analytical method

a) Fiber Stress at clamp.

![Diagram of a clamp with forces and calculations]

Gyrosopic force resultant on flywheel
shaft bags = 170 T

\[
\text{Force} = \frac{170}{2 \times \text{rad}(d)} \times \frac{90}{7.5} = 1032 \text{T}
\]

Take projected area of Hub as wobble load bearing area
(exremely conservative)

\[
A = 55.2 \times 7'' = 386''^2
\]

\[
\sigma = \frac{L}{A} = \frac{1032}{386} = 2.7 \frac{T}{\text{in}^2}
\]

Inner fiber stresses are so low even with this conservative estimate (probably less than 1 T/in²) that a more detailed analysis is not req'd.
b) **Deflection at Rim of Unstressed Cemented Flywheel**

For a conservative analysis, consider a strip twice the clamping hub width wide. Consider the shaft load applied at the radius of gyration as the gyroscopic effect depends on the energy level of the flywheel element rather than the centre of mass.

Consider the free deflection of each side (without the joint)

**Beam Moment (BM)**

\[ BM = \left(\frac{x-14.8}{2}\right) F \]

**Y'EI**

\[ Y'EI = \left(\frac{x-14.8}{2}\right)^2 F + C_1 \]

At \( x = 32.4'' \)

\[ Y'EI = \left(\frac{32.4-14.8}{2}\right)^2 180 = C_1 \]

\[ C_1 = -27900 \]

\[ Y'EI = \left(\frac{x-14.8}{2}\right)^2 180 - 27900 + C_2 \]
at \( x = 32.4 \) \( y = 0 \)

\[
\frac{\left(32.4 - 14.8\right)^3}{120} = 27,900 \times 32.4 + C_2 = 0
\]

\[
C_2 = 790,900
\]

at \( x = 0 \) \( \text{YEI} = C_2 = 790,900 \)

\[
y = \frac{740,400}{13,400 \times 3144} = 0.018''
\]

This indicates that the rim will not greatly deflect under the gyroscopic swashig load as the structure will be much more rigid than this simple model of it.

The gyroscopically induced stresses are seen to be an order of magnitude below those of the centrifugal variety. This would indicate at this early stage of analysis that they will not be particularly influential in the flywheel design.
Conclusions

1. Neither centrifugal loads nor gyroscopically induced swashing loads exceed the strength of readily available sheet steels.

2. The prestress will reduce these loads much further thus allowing the use of lower strength, more formable steels.

3. A more complete analysis is required to determine:
   a) Prestress stress distribution
   b) The actual effect of swashing forces (by finite element methods)

Though neither show any signs of causing any design or construction problems.
Addendum: Explanation of Prestress

The flywheel laminates are clamped around the hub as shown.

The lower clamping block does not come in contact with the laminates until the full prestress is achieved. The laminate centre hole is a close fit on the shaft. The clamped conical section of the laminates is slightly steeper so that as the clamping force F is applied it flattens out the clamped conical section of the laminate and also causes an equal and opposite reaction against the opposing laminates at the rim. This is represented by the following system:

As the radial load bearing area increases proportionally with radius, the radial stresses decrease linearly with radius.
The circumferential stresses are for the most part induced by the change in radius and therefore elongation of the circumferential elements.

\[
\epsilon_0 = \frac{(u+r) - r}{2\pi r}
\]

\[
\epsilon_0 = \frac{V}{r}
\]

These circumferential stresses will be compressive so long as \( \epsilon_0 < \pi r \)

where \( V \) = Poisson's ratio

At \( \epsilon_0 = \pi r \) there will be a nodal circle of zero circumferential stress.

At \( \epsilon_0 > \pi r \) the circumferential stresses will be tensile.

The flat rim behaves much as a thick walled cylinder and can therefore be treated as such. It will have a tensile circumferential stress which is largest at the interface between it and the shallow conical section.
"THE PRESTRESSED LAMINATED FLY-WHEEL AND ITS HYDROVAC AMBIENCE"

Robert C. Clerk,
20 Whitehill Road,
Glenrothes, Scotland.

ABSTRACT

The prestressed laminated flywheel seeks to retain the advantages of steel, in it low material cost, high specific inertia and low specific bulk, which serve to reduce ancillary weights of gearing and vacuum casings, while also extending the fatigue range and integrity beyond normal practice, so that when considered overall, the specific storage of the operating assemblage in Watt hours, is high.

The cold cross-rolled, low carbon steel sheets are pressformed as cymbal shaped "disc springs", assembled in opposed stacks enclasping the aluminium drive plate and spacer hub, and when pulled up by the tie rod and gripped by the stubshaft radial transfer concavities, thus prestresses in compression the critical central areas of the laminar assemblages. The peripheral nip on the low modulus drive plate ensures low centrifugal stress at the hub driving concentrarions. The Inverted stub-shaft journals counter normal production imbalance, have considerable effect on bearing power loss, noise and life expectancy, and like the support thrust bearing, are hydrostatic. The miniature vacuum/scavenge pump is hydraulic gear-motor driven, the casings are radially pleated to provide a baffled drainage sump, to increase thermal transfer area, and to reduce differential pressure deflection. The vacuum-stripped lubricant has a dissolved Helium residuum.

PREFACE

The history of flywheels and their usage up to 1963 has been embracingly covered in my SAE paper of that year. Until then, higher energy flywheels were forged weld-fabricated or cast, but we would no longer categorise these as high-energy.

Feasibility studies then had shown that the promise of highest energy per pound of flywheel weight lay with light weight filamentary constructions but a large part of this advantage would be lost to the larger and stiffer casing demanded by the greater bulk, and to the higher gear reduction to compensate for the increased speed. Also there appeared at the time to be a seemingly intractable problem at the driving transition between flywheel and supporting shaft; not to mention the cost aspect.

In the circumstances we decided to research ways of utilising the capabilities of steel more efficiently and in doing so conceived the pressed laminar construction with compressive prestress to bring the critical stress areas to an initial state of compression. We also discovered that the fatigue range and integrity of the steel sheets could be advantaged by cold reduction and cold-work forming and that fortuitously one of the lowest cost grades of steel was most responsive to this treatment.

FLYWHEEL DEVELOPMENT

Based on a long term background of unitary steel flywheels, extending from the war years, development of a laminated flywheel to the prestress theory progressed relatively quickly in the five defined areas:

(a) lamina contouring for optimized prestress gradient compromised to cold press forming and assembly stacking.

(b) application of prestress without unaccounted stress concentrations

(c) transmitting torque to and from the laminations without stress concentration.
(d) stability of the stacked assembly to
limiting thickness/diameter ratios
related to laminar vibration, gross
out-of-balance, precession and
journal vibrations

(e) optimising material specifications,
production forming and stacking of
laminations, and assembly and
balancing of complete flywheels.

PRINCIPLES OF CONSTRUCTION

Normally, even a pin-hole at the
center of a disc has a quite
disproportionate effect on centrifugal
stress, worsening progressively with
increasing hole diameter. Yet a
built-up construction, other than a
fully adhesive structure, demands
accommodation of at least a thru tie.

As the centrifugal tensile stress
approaching the center of a laminar disc
is approximately twice that at the
periphery, whereas the deflection stress
of an obtuse conical thin disc is
highly compressive near the centre,
balanced by mild tension at the
periphery, at approximately 8:1 ratio,
this forms the basis of our prestress
with opposed stacks of conical laminae
contra-loaded by a tie-rod of minimum
diameter, as Fig. 1.

ACCOMMODATING THE "SPINDLE":

To limit the applied preload, an
interstack axial spacer is required, and
to distribute the load application
demands a concave "collar" matching the
external convexity of the two outermost
laminae, the "collars" being adapted
as stub-shafts for supporting and
locating the flywheel and effectively
providing a built-up spindle.

LAMINAR STRESS DISTRIBUTION:

Both to offset the concentration
of stress around the central tie-rod
hole and to provide positive centering
of each stack of laminae, the central
area of each lamina is formed to a
very much more acute cone angle
(ca. 110° incl.) than the obtuse cone
angle (ca. 170° incl.) of main area
of the disc Fig. 2.

Prestressed Laminated Flywheel

FIG. 1
The tie-rod-load distributing concavity of each stub-shaft matches the acute conical area of the laminate very closely both in diameter and inclusive angle, as does the double-convexity of the inter-stack spacer, but their inclusive angle is fractionally greater and the spacer is short of the unloaded stacks by a predetermined gap which, when closed by the tie-rod load, limits the compressive prestress of the unclamped inner area to the desired figure. Any centrifugal loading must first neutralise this compression stress before going tensile to a fatigue-safe tensile limit.

The slight angular mismatch of the acute conical clamped area further increases prestress approaching the tie-rod hole and under centrifugal conditions there is considerable shedding of the radial/tangential loads to the reinforced stub-shaft collar such that critical tensile stresses around the hole are alleviated.

**TRANSMITTING DRIVE TORQUE**

On the face of it, the conical clamping to the stub-shaft should allow transmission of substantial inertia driving torque between the laminae and driving stub, especially if the laminae are appropriately spray bonded during assembly. However, a mathematical analysis shows that such torque transmission is very limited and any initiation of slip could disastrously affect balance and rigidity. Of course, any attempt to positively couple the clamped centres would generate stress concentrations as would negate the prestress inter-relationships.

As the greater inertia of the stacked laminae is effective at the periphery, and as the powerful deflection loading of the opposed stacks also interact at the periphery, by leaving the near-periphery of the laminae flat when press-forming, a driving plate can be securely clutched between the stacks at such a great torque radius that the possibility of slippage is non-existent. But at its centre the driving plate would have to be securely fixed to the clamping spacer which in turn must be torsionally located within the stub-clamped assembly.

Unfortunately, such a drive plate, if formed from steel sheet would be critically over-stressed at its drive centre. However, an aluminium alloy plate constrained to the same radial centrifugal strain expansion at its periphery as the steel laminae, will due to its very low Young's modulus be subject to a compressive rather than a tensile stress radially inwards of the clutched periphery, this compression reducing inwards to a nodal pitch circle which becomes the effective diameter for centrifugal stressing of the drive plate. This stressing is of course further minimised by the low specific weight of the aluminum alloy, such that liberties may be taken with the stress concentrations engendered by a secure driving connection to the clamping spacer.

At very high centrifugal loadings a single flat light alloy drive plate might be subject to compression buckling inwards of its periphery or radial creep of the clutched periphery, but this can be alleviated either by pressing wave rings around the compression area or by substituting slightly dished twin drive plates of lighter gauge.

The clamping spacer, which has now become a drive hub, must be torsionally located to the driving stub-shaft so that any incidental application of torque in excess of operational limits will not easily cause drive-hub slippage.

**LAMINAR STRESSES AND FATIGUE**

Actual laminar stress is engendered by the interaction of centrifugal (centripetal) forces with the assembly pre-stress. Fatigue is affected by integration of changes of tensile stress above the limit for infinite fatigue life and the number of times such changes occur. In a non-prestressed flywheel the greatest incidence of fatigue is that due to the flywheel running down to a standstill and subsequently accelerated back to
maximum operational speed. The energy differentials of normal operation will make a much smaller contribution to the total fatigue summation.

In a prestressed laminated flywheel, as the central areas which are subject to the greatest rate of change of stress will only be in fatigue effect when the rotation speed is exceeded beyond which centrifugal forces cancel the compressive prestress and exert themselves in tensile, only the peripheral area is subject to change of tensile stress over the entire speed range down to and upwards from standstill, and this at a low rate of change, unreversed and covering only a portion of the overall tensile stress. Consequently, as a general rule, the run down cycles can be neglected and the fatigue stress taken as effective only in operational differential energy cycling, the mean differential depth of each application determining the prestress necessary to effect fatigue balance between centre and periphery.

It is usual for flywheels to be used operationally in the upper two-thirds of their speed range, so utilising eight-ninths of their energy capacity. But in most applications the cycling mean is only 50% of this, the other 50% being reserved to cater for the less usual incidental demands.

It is generally considered appropriate to stress for infinite fatigue, but there will be applications where advantage can be taken of a low operational cycle count or a short life requirement by using very high tensile alloys (possibly of higher density) despite known deficiency in fatigue limits. However, it has been found that a low-cost low carbon steel (EN 2D/E) cold worked in finish rolling and press forming to around 116,000 lbf/sq.in. effective (125,000 max.) can have a fully reversed fatigue limit of 60,000 lbf/sq.in., better than most of the more costly high alloys and, due to the very low scatter in the formed laminae, safe at ± 47,000 lbf/sq.in.

For higher specific energies at infinite life, high-fatigue stainless steel (FV 520s) laminae can safely be stressed at ± 80,000 lbf/sq.in. equivalent to 16 Watt-hours/sq.in., as against 9 Wh/lb for the cold worked mild steel. Some later modern steel formulations, which have not yet been assessed at infinite life, offer even higher specific energies; and for finite limited life applications can give up to 50 Wh/lb without excessive drive speeds and with low specific bulk.

**STRESS METHOD**

For static prestress calculation we neglect the Almen-Les7io2 method as, for reasons later discussed, our chosen laminar thickness/diameter ratio will be negligibly minimal; but Hertzer's3 method is interesting in that it is concerned with the tensile fatigue aspect of the inner concave face resulting from deflection (our prestress), a condition only applying to our laminae at high centrifugal forces.

Although our prestress calculations are simplified by neglect of the stress differential between concave and convex faces of the laminae, they are complicated by the fact that deflection from static height is limited to the gently dished medial area, but the radial/tangential forces generated by deflection (Fig. 3) are accommodated additionally in the flat peripheral and the steeply coned central areas, the latter itself being subject to its own minimal clamped deflection and compressive load, and the former also sometimes being formed with a minimal dish so as to bear flat when subjected to the main deflection load.
DIMENSIONAL RELATIONSHIPS

Figure 2 shows typical dimensional relationships of a lamina, the frustro-conical height (h) being a compromise between the shallow height advisable to prevent tie-rod overstress at maximum laminar stacking and/or maximum laminar thickness (t. max.,) and the depth appropriate to ensuring adequate drive-plate peripheral nip and to obviate peripheral wave vibrations at minimum stacking.

PERIPHERAL DYNAMIC VIBRATIONS

A single lamina or an opposed pair of laminae at high rotational speed develop a peripheral wave the amplitude of which increases with rotational speed, and also decrease inversely as the cube of lamina thickness as has been shown reference Timoshenko4, Lamb5, Southwell6 and Sandor/Broniarek7. The Peripheral pre-tension of conically prestressed laminae therefore has some slight effect in delaying the onset and reducing the amplitude of peripheral vibration, but the effect of multiple stacking not only provides a stiffening inter-reinforcement but more importantly affords inter-laminar frictional damping or viscous-bond shear damping which effectively prevent the wave-form from developing to significant proportions at stacking levels above the currently accepted minimum of nine laminations in each demi-stack.

FLYWHEEL STABILITY

The structural stability of built up assemblies is always open to question, especially where they may be subjected to out-of-balance or elastic vibrations.

The prestressed laminated flywheel assembly owes its structural stability to two quite separate sets of forces, the laminar deflection loads counter-reacting at a large base-circle diameter, and the clamping loads active over the much smaller diameter of the stub-shaft clamping flanges but enjoying a stabilising augmentation from their deep-cone pressure angle.

At very high speeds the tendency towards "flattening" of the laminae reduces the wide-based counter-reactive loads. This is not sufficient to affect the EN 2D laminations seriously but must be taken into account in determining the additional prestress to be applied to FV.520(s) or other highly stressed laminae, especially where the maximum stacking limitation is approached, increasing the cumulative deflection load applied by the centre tie-rod whose stress may become critical unless laminar thickness is reduced.

The conically clamped central area is directly affected by rotational speed only to the extent that the centripetal radial/tangential forces shed from the laminations to the coned stub-flange may strain the flange rim sufficiently to reduce clamping nip at the flange periphery; but again this applies more to high-stress laminations than to EN 2D except at maximum stacking. However, the greater effective length between stub-flanges at maximum stacking affects stability in whirling (radial) deflection much more, varying as the cube of length in relation to the relatively small effective moment of inertia of the central clamped area.

FLYWHEEL FAILURE MODES

Flywheel failure will be due to one of two possible causes, an overstressed tie-rod or laminar crack propagation. Of these the former is less predictable, in absolute terms, as to the way its effects will act, but the bump/yump axial counterthrust would still maintain an overall hydraulic clamping force on a basically self-centring melange, and the relief of prestress would, at the more critical high speeds, be to some extent offset by centrifugal deflection of the laminae in the same sense as the original prestress.

Laminar failure, other than due to a flawed lamination escaping inspection will always commence with one of the outermost laminae.

The energy possessed by the largest likely detaching segment would suffice
at worst to puncture the casing without possibility of escaping, but the collapsed vacuum environment would have immediate effect to slow the flywheel rapidly to below sonic peripheral speed, more than halving centripetally induced stresses. The out-of-balance effect on the flywheel would also act to increase retarding drag. As a result the possibility of a catastrophic failure is negligible.

For some applications added assurance may be provided by adopting a higher fatigue material for the outermost laminations of each demi-stack.

MATERIALS AND FORMING METHODS

The preferred and most cost-effective material for lamination is EN 2D low-carbon steel work hardened by cold roll finishing in the strip mill to 215 Brinell (100,000 lbf/sq.in.) max., cut square and cross rolled in a reversing mill 50% total reduction to gauge size at 240 Brinell (110,000 lbf/sq.in.) max., centre-spot flame softened before blank-holder centre-punch pressed to form raising the near-centre worked area to 250 Brinell (116,000 lbf/sq.in.) max. An upsetrodking press is preferred as this allows the formed lamina to be slid on to a ball-ended transfer finger contacting only the centre hole so that a striker will produce a true ringing note if the lamina is without flaw. Although not entirely comparable, the fully documented work of Frost shows the very considerable effect of compressive (axial) and torsional cold working prestrain on the fatigue limit of En2C stressed in identical direction to prestrain as shown on Fig. 4. Also shown is the result of later work by Fox, Cartwright and Boxall of British Iron and Steel Research Association, where En2D strip has been rolled/compressed 50% of thickness before stressing in the orthogonal plane, producing a higher fatigue limit and greatly reduced tendency to crack propagation.

For higher energies, Firth-Vickers high fatigue stainless F.V. 520 (S) Steel in the Overaged 620 C condition exhibits the unusual characteristic that the fatigue limit (Fig. 5) coincides with the limit of proportionality and it's unaffected by any reasonable prehistory of excessive transgression, as may occur in the forming process. Although a considerable programme of explosive forming research was undertaken, it was not then practicable to assess possible damage to the fatigue characteristic due to shock, as has since been done for cold worked En2D which has been found to suffer quite severely as had been postulated. F.V.520 (S) is also less critical of the vagaries of powerspin forming than is En2D which latter would therefore appear to be at a lesser advantage for diameters beyond press-forming capability.
Likewise, development of the flywheel containment, bearings and other ambient conditions, from that used successfully in earlier forged flywheel applications, was concurrently rethought in six areas which might directly or indirectly affect basic flywheel development:

(a) casings stiffness and casting dies

(b) inner sheet-metal close-shrouding and sump-shield to reduce kinetic losses to lubricant and residual atmosphere

(c) hydraulic powered modular vacuum/savange mini-pump drawing down to low torr

(d) hydrostatic support thrust and inverted journal bearing to minimise bearing vibration losses

(e) input/output drive to the flywheel by gearing "invacuo" despite the difficult lubrication conditions

(f) a fully hermetic zero-ullage inclusive system permitting expansion of the pressurised "vacuum-stripped" working fluid

An exemplary arrangement of a hydro-kinetic accumulator Fig. 6 provides reference for discussion of these facets which are covered fully in my other paper presented in another session area.

As earlier mentioned, the prestressed laminar flywheel is a built-up assembly comprising, apart from the opposed laminar stacks, a drive-plate, centre spacer, stub-shafts and nutted tie-rod. It is sometimes advantageous to fill the inner voids of the flywheel with shaped cellular plastic elements as the air otherwise entrapped during assembly may later leak into the vacuum environment, upsetting the hermetic balance.

Rather than a single Aluminium alloy plate which can cause problems, we prefer to use twin lighter gauge driving plates, slightly coned and pre-indented for ball-indent drive which has proved far superior to peg-hole drives when subjected to reversing overloads.

The 3-piece construction adopted to accommodate the ball-indent drive will be no more costly to produce than a one-piece peg-hole drive hub if the shoulder spigots are eliminated by pre-broaching the centre element and by use of a sophisticated assembly jig. Also a Loctite fit of the coaxial convexed elements on the tie-rod spindle makes for a more rigid flywheel assembly.

Each stub-shaft fulfils four functions. Firstly, the concave flange serves as the external clamping surface locating a laminar stack. Secondly, the stiffened flange periphery accepts the transferred radial component of centripetal forces acting on the laminar stack. Thirdly, the cupped outer end of the stub serves as the rotating outer race of an inverted journal bearing locating the flywheel axis. The fourth function is different for lower and upper stub-shafts, the lower cup-lip supporting the vertical-axis flywheel on a foot-step bearing, whereas the upper stub-cup carries the drive-transmitting gear.

Splined for torsional location of the drive-hub and stub-shafts, the tie-rod can be very highly stressed in tension as it is not fatigue conscious in this aspect but it must be fully creep resistant.

For very high energies, the F.V. 520(S) laminar stacks will be plain broached at full deflection to a close fit on the tie-rod, as will the clamping members, with liberal use of anaerobic
Loctite at assembly. For some applications the splining may be deleted in a fully adhesive assembly, but the transmitted torque must be strictly limited.

FLYWHEEL BEARINGS

The high running speed, and precession forces due to vehicle movement, did not prove to be the most significant bearing problems; rather those due to excess lubrication and to flywheel out-of-balance effects. The lubrication problem could have been handled by a special high-angle-contact ball footstep bearing combining journal and thrust support, but the out-of-balance precession caused the bearing load to advance around the running clearance of the outer race, whether this was static as in the case of the footstep bearing or rotating at equal speed as was the upper pilot bush in the clutches bevel-drive pinion, which suffered more when clutched than when running free in a static pinion bush.

INVERTED JOURNALS

At high running speeds the vibration precessions transmit considerable amounts of energy into noise and heat however "perfect" the balance and however close the original running clearance.

A study of the forces involved showed that the out-of-balance precession effect could effectively be nullified if the bearing configuration were inverted; that is, the outer race or bush revolved, with the flywheel, about the fixed journal pin locating the force reactions. A roller journal was ruled out by lubrication problems and a ball bearing generated and transmitted too much noise, even when mounted in resilient sleeves.

HYDRODYNAMIC BEARINGS

Although solving the noise problem, hydro-dynamic journal bearings were subject to shear-drag losses which appeared to be quite intractable even in the inverted configuration with its mildly regressive load angle. There appeared to be no "right" position to introduce the lubricant as the dynamic pressure wedge kept moving around and pumping back with drastic collapse.

RHEODYNAMIC JOURNALS

The application of hydrostatic pressure areas to the static stub "pin" of the inverted journal at least gave predictable and reliable results, with even lower noise transmission. But at efficient "leakage" flows the shear-drag loss was higher than was acceptable for a "total energy" system.

Two avenues of shear reduction were explored. First, as shear drag varied proportionally with differential surface speed, a half-speed floating bush carrying subjoined pressure areas on its outer surface was found to halve the drag. Secondly, boundary land clearances were tailored inversely to the pressure differentials across them and this had a dramatic effect on the shear drag, though with more deterioration of frequency response "stiffness" than would have been acceptable at the highest running speed. However this was easily corrected by retailoring the clearance inverse function.

Eventually both ploys were used in combination to good effect; and even the incorporation of both sets of pressure areas within and without the annular floating sleeve appears to make only a very small and quite acceptable difference to the response definition.

RHEODYNAMIC FOOTSTEP SUPPORT

A flat annular hydrostatic pressure area bounded by inner and outer lands provides a simple and effective supporting thrust bearing, with adequate response to high vehicle bump acceleration but lacking stability in rebound or zero-G jump conditions.

As a countermacting thrust of similar construction acting from above the flywheel unit is impracticable for a number of reasons, a non-dimensional hydraulic thrust equivalent to approximately 50% of flywheel weight is applied to stabilise the down-load during rebound or similar conditions.
FOOTSTEP THRUST WASHER

When the flywheel is going out of service its speed will run down to standstill. But before it runs right down, the footstep thrust supporting pressure will have reduced below the critical minimum for the bearing load, resulting in scored bearing faces. To obviate this a thrust washer is introduced having on one face the annular pocket and boundary lands for the hydrostatic support pressure, and on the other face Glacier DX anti-friction facing material which will provide the running surface at the low-run down speeds. It also has a toothed or vaned periphery which fulfils another function later described.

FLYDRAULIC PARASITIC ENERGY/LOSS.

Until magnetic thrust support and journal location are commercially feasible, flywheel ambience must be effected by a vacuum/scavenge ancillary pump with an energy-loss penalty substantially proportional to the through-flow of bearing effluent. Nevertheless this becomes a secondary priority in comparison with bearing losses related to lubricant through-flow, and even to driving gear lubrication and pump/motor leakage.

Taking first only the flywheel related losses, the ancillary pressure flow required to "lubricate" the hydrostatic bearings and to power the vacuum/scavenge pumping are related to flywheel size and weight and therefore a constant, whereas the bearing shea-r-drag losses vary with speed, as do the windage losses unless these are maintained low enough to be neglected.

RHEODYNAMIC BEARING LOSSES (SUPPORT)

Fluid film shear losses of rheodynamic thrust support bearings are a function of supported weight, bearing diameter, responsive land ratio, pressure drop and rotation speed. The pressure drop is effectively fixed by the designed level of ancillaries pressure in absolute units (Pabs = 520 p.s.i.a.); the land ratio should depend upon the application vertical acceleration characteristic, but in the interests of standardisation we work at 40%: the bearing diameters are determined, the inner by the journal pin (10% of flywheel diameter; see below) and the outer by the support area at 80% flotation, 1.5 g vertical bump acceleration plus 0.5 Mg counterthrust, giving an area equivalent to 3.75 Mg/Pabs, at a pre-impedenced normal support pressure of 0.5 Pabs for Mg maximum and higher impedences for lower flywheel weights.

Although the speed factor could be taken at the mean of the flywheel 3:1 speed range, the losses are more critical at maximum storage speed, and on the above basis the energy loss from this bearing source will approximate to 0.45% of max. energy.

RHEODYNAMIC BEARING LOSSES (JOURNAL)

The 3-pad journal bearings (one above and one below the flywheel) are determined sizewise by their intercept distance, by the allowable out-of-balance of production flywheels, by the operating speed, and by upset precession forces.

The inverted bearing construction has the internal bore of each cupped flywheel stubs-shaft rotating about a fixed journal pin through the intermediary of a "half-speed" bearing sleeve in which are formed the hydrostatic pad recesses fed by three radial holes in the journal pin, each separately impedance from the ancillary pressure source.

This system is not ideal as compared with pad recesses formed around the journal pin but is adequate and has production advantages. The "half-speed" sleeve effectively halves the speed related shear drag which is further reducing by increasing the clearances at the inter-pad lands. The combined losses of the upper and lower journals at maximum storage speed approximate to 0.65% of gross energy for shear and leakage, at the present state of development.
VACUUM/SCAVENGE PUMPING LOSS

To scavenge the bearing leakages and to maintain a 10⁻¹ torr vacuum requires 0.35% of gross which added to the bearing losses gives a total of 1.45% of gross energy. Therefore a 10 kWh flywheel would require 0.145 kW to keep it at maximum energy, exclusive of flywheel windage which, by using residual Helium in a cost-effective level of vacuum ambience, appears to be negligible.

FLYWHEEL CASINGS AND ANCILLARIES

The flywheel casings carry the flywheel journal and support bearing and provide an enclosed vacuum environment for the flywheel, a drainage sump for bearing effluent and pump leakage oil awaiting scavenge clearance, a location for the drive gearing, a mounting face for the hydraulic pump/motor, and finally a bottom to the oil reservoir.

THE LOWER (SUMP) CASING:

The relatively large diameter exposes the evacuated casings to a considerable atmospheric pressure loading, necessitating a stiff construction to avoid excessive axial strain deflection. Early casings had radial rib reinforcements but nevertheless were subject to excessive deflection which caused problems. A conical formation with inward projecting deep radial folds has since proved adequately stiff, reduced the internal cone volume, provided anti-surge baffles for any temporarily unsavenged sump drainage, and increased the external direct cooling surface area. This axial pressure/vacuum loading provides a uniformly distributed load at the joint-face requiring only a circlip for its location. The fixed journal bearing pin carrier and the vacuum/scavenge motor/pump modular assembly are separately inserted in the casings, and cast-in galleries service them with input power fluid and scavenge outlet.

UPPER (DRIVE-SIDE) CASING

The upper casing also serves as the bottom of the pressurised oil reservoir and will require to be adequately reinforced against collapse deflection. It carries the flywheel upper-bearing, spigot for the output/input pump/motor, and bearing housings for the drive gearing, symmetrically disposed to accept one or more pump/motor units to allow of greater power output or a multi-plexed circuit.

Cast-in galleries service the flywheel upper bearing and rebound stabiliser, the drive gear bearings, the gear tooth retreating spray, and if required a quill-driven hydraulic speed-signal generator for energy-level indication, duplicating that incorporated in the hydraulic pump/motor.

DRIP AND SURGE SHIELDS

As waste oil from the flywheel upper bearing and the drive gearing coming into contact with the flywheel would acquire kinetic energy and dissipate it as heat to the outer casing, a light-gauge pressed sheet drip shield is incorporated to deflect such waste oil, as well as the pump/motor internal leakage, to the sump scavenge.

Another shield closely surrounds the flywheel periphery and underface and the lower stub-shaft to protect against surge of any unsavenged oil temporarily resident in the sump, despite the sump baffles. It also shrouds the scavenge primer intake.

VACUUM SCAVENGE PUMP MODULE

The gear-motor driven vacuum/scavenge/mini-pump withdraws the sump oil and maintains a near absolute vacuum (10⁻² torr) returning the extravasation to the hermetically pressurised oil reservoir. It is of the root-supercharged helical gear type, centrifugally primed by the vanned thrust bearing washer, and is combined with the gear-motor as a replaceable modular insert communicating with the ancillary pressure supply and reservoir return galleries in the flywheel casing.

WORKING FLUID

The advisability of using a vacuum-striped oil has been mentioned earlier, for reasons by now apparent.
However other aspects of the oil specification benefit from the fully hermetic system with no requirement for additives directed to anti-foaming, anti-oxidation, anti-acid, anti-sludging, anti-thermal cracking and other "antis", but a measure of dissolved Helium and constancy of viscosity index between say - 20° to + 70° Centigrade, approximating to 45 micro-reyns at 60°C. The residual Helium allows of a cost-effective level of vacuum (ca 10⁻¹ torr) without fear of Hydrogen embrittlement.

FLYWHEEL/PUMP CONNECTING DRIVE

Early Hydraulic accumulator layouts incorporated a disconnecting drive, as the idle-running drag loss of the best available pump/motor was so excessive that the stored inertia energy would have been quickly dissipated. However, the pump/motor researched and developed over the past eight years has exhibited such an extremely low idle-running drag loss (excluding the proprietary miniature gear pump, temporarily used for priming the controls, which at present trebles the drag loss) that it is now practicable for the pump/motor to be direct connected. However, "direct connected" to a coaxially disposed pump/motor would mean speed-matching a fairly big flywheel to a rather small pump/motor without alternative unless a planetary drive intervened.

But an offset gear drive offers several important advantages. First a substantial reduction in overall (vertical) height. Secondly, the gearing ratio is easily altered to match any size (or speed) flywheel to any size (or speed) pump/motor. Thirdly, the flywheel stub-shaft gear pinion can drive, not just the one but up to five not necessarily identical pumps separately circuited to power and control a number of individual machines or services from a single energy storage source.

The pump/motor shaft is connected to the driving gear-wheel by a sleeve quill so that the pump shaft can bend quite freely without affecting the drive gearing.

Gear lubrication is by cooling jets spraying the gear teeth immediately post-engagement, except for very high peripheral speeds at high power throughput, when near total fling may necessitate a finely atomised spray of lubricant to the pre-engagement faces, proportioned to pump/motor pressure, as there will be no floating oil mist in the vacuum environment.

FLYWHEEL ENERGY CAPACITY

In any application of the flywheel, possibly the prime determinant will be energy capacity to be provided. Almost certainly, there will be limitations in one or more directions prescribed by manufacturing, assembly, power transmission, installation, servicing, life expectancy or for a dozen other reasons.

FLYWHEEL SIZE DETERMINATION

Usually, the cyclic energy differential of a dynamic system divided by the power transmission efficiency plus a percentage reserve according to application establishes the energy capacity. Thus in the case of a motor vehicle the cycle to be considered is acceleration to maximum or operational speed and subsequent retardation to standstill; the efficiency say 90%; and to cater for the vehicle potential energy gained or lost up and down hills and inclines, say 100% reserve; these totalled must be equated to 8/9ths of flywheel energy capacity as the flywheel is operated exclusively in the upper two-thirds of its speed range.

Installation confines will determine thickness/diameter ratio in association with the power transmission ancillaries.

PRESS-TOOL PRESCRIBED SIZE RANGE:

Press-formed flywheel laminations demand a costly combination press-tool which would suggest a diametrically stepped range of flywheels, each covering a wide energy range by multi-stacking variations, though minimum stacking makes inefficient use of the common components such as drive plates, stub-shafts and casings.
The range tentatively prescribed starts at 18 inches, progressing in 6 inch increments to 88 inches which is the largest diameter presently considered for press-forming. Beyond this progression would be by 12 inch steps, and forming by power-spinning or explosive forming. Table A provides a guide to flywheel size factors.

Table A. Laminar Flywheel Range - Laminations Formed From Low Carbon Steel

<table>
<thead>
<tr>
<th>Diam.</th>
<th>Max. Gauge</th>
<th>Weight(lb)</th>
<th>Drive Centre (lb)</th>
<th>R.P.M. Max.</th>
<th>kWh/1&quot; Thick</th>
<th>Centre kWh</th>
<th>Stacked Laminar Thickness</th>
</tr>
</thead>
<tbody>
<tr>
<td>18&quot;</td>
<td>.040&quot;</td>
<td>72</td>
<td>10</td>
<td>21,000</td>
<td>0.575</td>
<td>0.0216</td>
<td>0.75&quot;/3.5&quot;</td>
</tr>
<tr>
<td>24&quot;</td>
<td>.056&quot;</td>
<td>128</td>
<td>25</td>
<td>15,750</td>
<td>1.02</td>
<td>0.0384</td>
<td>1.0&quot;/5&quot;</td>
</tr>
<tr>
<td>30&quot;</td>
<td>.064&quot;</td>
<td>200</td>
<td>45</td>
<td>12,600</td>
<td>1.60</td>
<td>0.060</td>
<td>1.25&quot;/6.5&quot;</td>
</tr>
<tr>
<td>36&quot;</td>
<td>.080&quot;</td>
<td>288</td>
<td>80</td>
<td>10,500</td>
<td>2.3</td>
<td>0.0865</td>
<td>1.5&quot;/7.5&quot;</td>
</tr>
<tr>
<td>42&quot;</td>
<td>.092&quot;</td>
<td>392</td>
<td>126</td>
<td>9,000</td>
<td>3.13</td>
<td>0.118</td>
<td>1.75&quot;/9&quot;</td>
</tr>
<tr>
<td>48&quot;</td>
<td>.104&quot;</td>
<td>512</td>
<td>187</td>
<td>7,875</td>
<td>4.09</td>
<td>0.154</td>
<td>2&quot;/10&quot;</td>
</tr>
<tr>
<td>60&quot;</td>
<td>.128&quot;</td>
<td>800</td>
<td>360</td>
<td>6,300</td>
<td>6.39</td>
<td>0.240</td>
<td>2.5&quot;/12.5&quot;</td>
</tr>
<tr>
<td>72&quot;</td>
<td>.160&quot;</td>
<td>1152</td>
<td>645</td>
<td>5,250</td>
<td>9.2</td>
<td>0.345</td>
<td>3&quot;/15&quot;</td>
</tr>
<tr>
<td>84&quot;</td>
<td>.192&quot;</td>
<td>1568</td>
<td>1030</td>
<td>4,500</td>
<td>12.5</td>
<td>0.47</td>
<td>3.5&quot;/17.5&quot;</td>
</tr>
<tr>
<td>96&quot;</td>
<td>.212&quot;</td>
<td>2048</td>
<td>1525</td>
<td>3,940</td>
<td>16.4</td>
<td>0.628</td>
<td>4&quot;/20&quot;</td>
</tr>
<tr>
<td>108&quot;</td>
<td>.232&quot;</td>
<td>2592</td>
<td>2100</td>
<td>3,500</td>
<td>20.7</td>
<td>0.79</td>
<td>4.5&quot;/22.5&quot;</td>
</tr>
<tr>
<td>120&quot;</td>
<td>.252&quot;</td>
<td>3200</td>
<td>2820</td>
<td>3,150</td>
<td>25.6</td>
<td>0.96</td>
<td>5&quot;/25&quot;</td>
</tr>
<tr>
<td>144&quot;</td>
<td>.324&quot;</td>
<td>4408</td>
<td>4815</td>
<td>2,625</td>
<td>36.8</td>
<td>1.38</td>
<td>6&quot;/30&quot;</td>
</tr>
<tr>
<td>168&quot;</td>
<td>.375&quot;</td>
<td>6272</td>
<td>8000</td>
<td>2,250</td>
<td>50.0</td>
<td>1.88</td>
<td>7&quot;/35&quot;</td>
</tr>
<tr>
<td>197&quot;</td>
<td>.433&quot;</td>
<td>8630</td>
<td>11000</td>
<td>1,920</td>
<td>69.0</td>
<td>2.95</td>
<td>8.2&quot;/41&quot;</td>
</tr>
</tbody>
</table>

NB Gross Weight of Accumulator Assy. (less Pump & Oil) = Weight of Flywheel + 25 Laminae (Cast Iron Casings) or + 10 Laminae (Al. Alloy Casings).

FV 520(s) Laminations will run 35% Faster and Store 82% More Energy
FLYWHEEL APPLICATIONS

The prime application envisaged for this project has always been related to vehicle propulsion systems, although this has not prevented us from studying other application areas, some in quite considerable detail.

Even before our earlier work on the Gyreacta flywheel/mechanical transmission, our 1949 studies showed that the optimum flywheel power transmission should be hydraulic, except that we could nowhere find a suitable pump/motor complementary in characteristics to the flywheel. Consequent to the 1963 SAE paper, market research showed the need for a high energy hydraulic accumulator but we still could not find a pump/motor with performance and control characteristics remotely approaching the flywheel requirement.

Further market research in 1968 showed that a successful hydrostatic automotive transmission would demand a pump/motor and control system very similar in overall characteristic to that for the high-energy accumulator; so fundamental study, research and development programmes were instituted the results of which are chronicled in a paper presented in another session of this symposium.

The union of the developed pump/motor and the laminated flywheel as a high-energy accumulator is covered in session paper earlier mentioned and illustrated as Fig. 6.

References

APPENDIX 1
PRESSURE VESSEL DESIGN PHILOSOPHY

Design would be in accordance with BSS 5500: 1976 Category 1 and latest Amendments.

The scantlings would be calculated for a design internal pressure of 5 Hg abs. and external pressure of 5 bar, a minimum metal temperature of -10°C and a maximum metal temperature of 50°C.

The scantlings would be checked for an emergency external pressure of 10 bar.

The vessel would be designed for a working life of 25 years, being subjected to sea 'G' loadings, and fluctuating gyroscope loadings.

The vessel shell would be suitably locally increased in thickness to withstand the loadings from the gyroscope and generator supports.

The butt welded seams would be subjected to 100% radiography.

All fillet welds subject to cyclic loading conditions would be ground to a smooth radius of the toes of the fillet welds.

All fillet weld attachments would be crack detected.

The material for the vessel would be a fine grained carbon manganese steel and meet the requirements of BSS 5500 for Low Ambient Temperature Service.

The vessel would be stress relieved as required to meet the requirements of BSS 5500 for Low Ambient Temperature Service and cyclic loading conditions.
EDINBURGH—SCOPE—LAING

LAING

5TH YEAR

WAVE ENERGY

REPORT

APPENDIX 2

CIVIL DESIGN PHILOSOPHY AND CALCULATIONS
EDINBURGH WAVE POWER PROJECT

REVIEW 1979

CIVIL ENGINEERING COSTS INPUT

John Leing Construction Limited
Newton House, Charing Cross
477 Sauchiehall Street
Glasgow G2 3NJ

041-331 2731 : Telex 779882
PREAMBLE

This civil engineering costs input comprises short bills of quantities which have been priced at cost levels applicable in October 1979. The effect of future cost fluctuations has not been taken into account. Rates in the bills for the more significant items are supported by calculations. Comparisons with current costs for other civil engineering works are also included.

The bills of quantities relate to :-

- The manufacture, launching and mating of 1020 prestressed concrete duck bodies and 510 prestressed concrete duck spines at an unidentified greenfield site on Scotland's west coast.

The pricing relates to sequences and methods of construction which are, briefly :-

**Bodies**:
Beaks would be delivered to site by sea. In turn each beak would be placed on end in a body casting location and provide permanent formwork to the front face of a body. The body complete with diaphragms would be cast in four lifts. After 'single end' post tensioning the body complete with beak would be launched, turned and mated to a spine. Output - One every two days.

**Spines**:
These would be made in seven precast reinforced concrete segments cast on end. The segments would be turned through 90° and jointed with six 100 mm wide infills of rapid hardening grout. After the positioning of two male or two female jointing units each spine would be post tensioned. The jointing units would be secured by the tensioned cables and anchors. Each complete spine would then be launched and mated to two bodies. Output - One every four days.
PREAMBLE (continued)

Programme

Basically 1020 bodies and 510 spines in 10 years.

Allow 9 months for yard construction, ie from initial entry to greenfield site to commencement of body and spine manufacture.

\[10 \text{ years} - 0.75 \text{ yr} = 9.25 \text{ years}\]

After allowance for holidays (CE Working Rule Agreement 1980 onwards) each year will comprise approximately 48 weeks of 5 days, ie 240 working days per annum.

\[9.25 \text{ years} \times 240 \text{ wkg days} = 2220 \text{ working days}\]

Bodies: \[2220 \text{ wkg days} \div 1020 \text{ bodies} = 2.18 \text{ wkg days}\]

Output 1 body every 2 days (\(\vdash\) allowance for slippage approximately 10%)

3 lines of 7 casting positions = 21 total. Cycle time for use of each position 35-42 days
(42 days \div 21 positions = 2 days)

Spines: \[2220 \text{ wkg days} \div 510 \text{ spines} = 4.35 \text{ wkg days}\]

Output 1 spine every 4 days (\(\vdash\) allowance for slippage approximately 10%)

3 lines of 7 casting beds - each line producing a spine of 7 segments in 12 working days
EWPP : REVIEW 1979 : CE COSTS INPUT

SPINES : Per Spine

Note: The quantities below are averages of a male spine and a female spine.

<table>
<thead>
<tr>
<th>Item</th>
<th>Description</th>
<th>Qty</th>
<th>Rate</th>
<th>£</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Formwork, circular, to inner and outer surfaces</td>
<td>3550 M2</td>
<td>7.91</td>
<td>28,080.50</td>
</tr>
<tr>
<td>2.</td>
<td>Reinforcement - High yield</td>
<td>127 Mg</td>
<td>328.77</td>
<td>41,753.79</td>
</tr>
<tr>
<td>3.</td>
<td>Reinforced concrete</td>
<td>1265 M3</td>
<td>30.67</td>
<td>38,797.55</td>
</tr>
<tr>
<td>4.</td>
<td>Prestressing</td>
<td>Item</td>
<td></td>
<td>77,787.00</td>
</tr>
<tr>
<td>5.</td>
<td>Buoyancy tanks - concrete pipes</td>
<td>Item</td>
<td></td>
<td>6,000.00</td>
</tr>
<tr>
<td>6.</td>
<td>Take delivery of end units and place in position before stressing</td>
<td>2 No</td>
<td>202.50</td>
<td>405.00</td>
</tr>
<tr>
<td>7.</td>
<td>Launch</td>
<td>Item</td>
<td></td>
<td>607.50</td>
</tr>
<tr>
<td>8.</td>
<td>Mating to bodies (see item B1/11)</td>
<td>Item</td>
<td></td>
<td>Included</td>
</tr>
<tr>
<td>9.</td>
<td>Provisional Sum: Allow for sundry items which would be measured separately in a complete bill of quantities</td>
<td>Item</td>
<td></td>
<td>24,000.00</td>
</tr>
</tbody>
</table>

Total 217,431.34

John Laing Construction Limited
<table>
<thead>
<tr>
<th>Description</th>
<th>Rate (No)</th>
<th>Rate (£m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Duck Bodies</td>
<td>1020</td>
<td>68.3</td>
</tr>
<tr>
<td>From Page B1</td>
<td></td>
<td>69.7</td>
</tr>
<tr>
<td>2. Spines</td>
<td>510</td>
<td>217.4</td>
</tr>
<tr>
<td>From Page B2</td>
<td></td>
<td>110.9</td>
</tr>
<tr>
<td>3. Preliminaries</td>
<td></td>
<td>54.1</td>
</tr>
<tr>
<td>From Page C3</td>
<td>Item</td>
<td></td>
</tr>
<tr>
<td>4. Design Fees, Overheads and Profit</td>
<td>Item</td>
<td>35.9</td>
</tr>
</tbody>
</table>

Civil Engineering Total: 270.6

John Laing Construction Limited
### DUCK BODIES: Per Body

<table>
<thead>
<tr>
<th>Item</th>
<th>Qty</th>
<th>Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Take delivery of beak unit, place in position on end to provide permanent formwork to front face of body</td>
<td>Item</td>
<td>162</td>
</tr>
<tr>
<td>2. Formwork, circular, to vertical surfaces</td>
<td>813 M²</td>
<td>8.33 £</td>
</tr>
<tr>
<td>3. Formwork to voids - vertical</td>
<td>980 M²</td>
<td>12.18 £</td>
</tr>
<tr>
<td>4. Formwork to voids - horizontal</td>
<td>46 M²</td>
<td>11.19 £</td>
</tr>
<tr>
<td>5. Reinforcement - High yield</td>
<td>41 Mg</td>
<td>328.77 £</td>
</tr>
<tr>
<td>6. Lightweight reinforced concrete</td>
<td>337 M³</td>
<td>47.94 £</td>
</tr>
<tr>
<td>7. Prestressing</td>
<td>Item</td>
<td>7124.75</td>
</tr>
<tr>
<td>8. Formwork to void - vertical</td>
<td>57 M²</td>
<td>12.18 £</td>
</tr>
<tr>
<td>9. Mass concrete</td>
<td>30 M³</td>
<td>2616 £</td>
</tr>
<tr>
<td>10. Launch body complete with beak</td>
<td>Item</td>
<td>405.10</td>
</tr>
<tr>
<td>11. Mate body complete with beak to spine - floating</td>
<td>Item</td>
<td>1250.10</td>
</tr>
<tr>
<td>12. Provisional Sum: Allow for sundry items which would be measured separately in a complete bill of quantities</td>
<td>Item</td>
<td>9000.00</td>
</tr>
</tbody>
</table>

**Note:** The words 'vertical' and 'horizontal' above relate to the body in its cast on end attitude

---

**Total** 68278.84

John Laing Construction Limited
<table>
<thead>
<tr>
<th>Item</th>
<th>Qty</th>
<th>Rate</th>
<th>£000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acquisition of site</td>
<td>1</td>
<td>40.00</td>
<td></td>
</tr>
<tr>
<td>Cranage etc.</td>
<td>42</td>
<td>2.50</td>
<td>105.00</td>
</tr>
<tr>
<td>Spine rotating units</td>
<td>2</td>
<td>400.00</td>
<td>400.00</td>
</tr>
<tr>
<td>Spine stressing cradles</td>
<td>42</td>
<td>2.50</td>
<td>105.00</td>
</tr>
<tr>
<td>Gantry cranes: 1250T</td>
<td>2</td>
<td>5,000.00</td>
<td>5,000.00</td>
</tr>
<tr>
<td>: 650T</td>
<td>2</td>
<td>3,000.00</td>
<td>3,000.00</td>
</tr>
<tr>
<td>: 50T</td>
<td>6</td>
<td>600.00</td>
<td>1,800.00</td>
</tr>
<tr>
<td>Transferer</td>
<td>2</td>
<td>750.00</td>
<td>750.00</td>
</tr>
<tr>
<td>Synchrolift</td>
<td>2</td>
<td>1,100.00</td>
<td>1,100.00</td>
</tr>
<tr>
<td>Operators</td>
<td></td>
<td></td>
<td>1,633.90</td>
</tr>
<tr>
<td>Fuel</td>
<td></td>
<td></td>
<td>800.00</td>
</tr>
<tr>
<td>Upkeep</td>
<td></td>
<td></td>
<td>1,458.60</td>
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Total 16,087.80

John Laing Construction Limited
### PRELIMINARIES (Continued)

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<th>Qty</th>
<th>Rate</th>
<th>£000's</th>
</tr>
</thead>
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<tr>
<td>3</td>
<td>Buildings</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3.1</td>
<td>Covered working area (factory conditions) over concrete casting, stressing &amp; spine buffer storage areas</td>
<td>44,800 M2</td>
<td>£2.11</td>
<td>94,528</td>
</tr>
<tr>
<td>3.2</td>
<td>Yard areas and roads - concrete</td>
<td>7050 M2</td>
<td>£0.27</td>
<td>190.4</td>
</tr>
<tr>
<td>3.3</td>
<td>Workshops</td>
<td>300 M2</td>
<td>£3.20</td>
<td>96.0</td>
</tr>
<tr>
<td>3.4</td>
<td>Offices, Canteens, Stores and the like</td>
<td>3100 M2</td>
<td>£1.48</td>
<td>439.6</td>
</tr>
<tr>
<td>3.5</td>
<td>Upkeep</td>
<td>Item</td>
<td></td>
<td>763.4</td>
</tr>
<tr>
<td>4</td>
<td>Staff; Salaries and all employment costs</td>
<td>Item</td>
<td></td>
<td>115,660.0</td>
</tr>
<tr>
<td>5</td>
<td>Provisional Sum: to allow for Attendance, Office Cleaning, Camp/Caravan Site, First Aid, Security, Setting Out Instruments, Laboratory, Rates, Water, Electricity (except for plant), Signs, Protection of Public Roads and Existing Services, Site Transport, Bonds, Insurance, Progress Photographs, Petty Cash, Postage, Stationery, Copying, Computer Charges and Sundries including Clear Site on completion</td>
<td>Item</td>
<td></td>
<td>11,377.8</td>
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</table>

Total: £37,980.0

John Laing Construction Limited
<table>
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<th>Description</th>
<th>Qty</th>
<th>Rate</th>
<th>£000's</th>
</tr>
</thead>
<tbody>
<tr>
<td>From Page C1</td>
<td></td>
<td></td>
<td>16,087.8</td>
</tr>
<tr>
<td>From Page C2</td>
<td></td>
<td></td>
<td>37,980.0</td>
</tr>
<tr>
<td><strong>CE Preliminaries Total</strong></td>
<td></td>
<td></td>
<td>54,067.8</td>
</tr>
</tbody>
</table>
COST COMPARISONS

Gross cost per cubic metre of concrete

Quantity of concrete

\[
\begin{align*}
1020 \text{ Duck Bodies} & \times 367 \text{ m}^3 = 374,340 \text{ m}^3 \\
510 \text{ Spines} & \times 1265 \text{ m}^3 = 645,150 \\
\hline
\end{align*}
\]

\[1,019,490 \text{ m}^3 \]

\[
\begin{align*}
\frac{\£270,600,000}{1,019,490 \text{ m}^3} &= \£265.43 \text{ per m}^3
\end{align*}
\]

Accepted tender gross costs per cubic metre of concrete

- Reinforced concrete raw water tank - West Central Scotland 1978: £210.00
- Large reinforced concrete sewage treatment works - Northern England 1979: £205.00
- Reinforced concrete water treatment works - Northern England 1978: £214.00
- Reinforced concrete water tower - East Anglia 1979: £235.00
- Post tensioned concrete sugar silo - East Anglia 1979: £410.00
- Major Colliery Foundation Works - North Yorkshire 1979: £549.00
- River Bridge of 204m main span - Northern Ireland: £660.00
<table>
<thead>
<tr>
<th>Bill Ref</th>
<th>Quantity</th>
<th>Unit</th>
<th>DETAILS</th>
<th>Rate</th>
<th>A-Labour</th>
<th>B-Plant</th>
<th>C-Material</th>
<th>D-Durable</th>
<th>E-Durable</th>
<th>F-Expense</th>
<th>H-I</th>
</tr>
</thead>
<tbody>
<tr>
<td>B1/1</td>
<td>Item</td>
<td></td>
<td>Take delivery of beam unit, place in position on end to provide permanent formwork to front face of body - WT 464 Tonnes</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Crampage etc in prelims</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Lab attend</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>4 men x 10hrs @ 4.05</td>
<td></td>
<td>162.00</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>


<table>
<thead>
<tr>
<th>Bill Ref.</th>
<th>Quantity</th>
<th>Unit</th>
<th>DETAILS</th>
<th>Rate</th>
<th>A-Labour</th>
<th>B-Plant</th>
<th>C-Material</th>
<th>D-Durable</th>
<th>E-Durable</th>
<th>F-Expense</th>
<th>H:</th>
</tr>
</thead>
<tbody>
<tr>
<td>B1/2</td>
<td>813</td>
<td>M2</td>
<td>Formwork, circular, to vertical surfaces</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Make: Bt in steel moulds</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.87</td>
<td>0.09</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Current price (flat) $147/m²</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>x 1.3 = $191-10/m²</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$191-10 per m² ÷ 102 uses (10 sets total)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>9.85</td>
<td></td>
<td>6.6</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Maintenance 5% of lost</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1.87</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fix &amp; strike:</td>
<td>1.3h @ 3.73 Tradesman</td>
<td>9.85</td>
<td>1.3h @ 3-05 lab + 6 (7:6 ratio 6:1)</td>
<td></td>
<td></td>
<td></td>
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Note: Calculations and rates provided.
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\[ 7124 \times 1332 = 400 \times 5392 \]
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Say as 81/3
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<td>Coarse 1.126 T @ 4-15 Nett + 2 1/2% waste</td>
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<td>Item</td>
<td>Tapered body complete with boss, Coreage, Traverse etc &amp; Prells, Lab able.</td>
<td>Rate</td>
<td>A-Labour</td>
<td>B-Plant</td>
<td>C-Material</td>
<td>D-Durable</td>
<td>E-Durable</td>
<td>F-Expense</td>
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<td>Equipment etc in Prelims</td>
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<td>10 men x 20hrs @ 4.05</td>
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<td>Divers 3 men team</td>
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<td>Supervisor + 2 divers</td>
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<td>£220 per 8 hrs</td>
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<td>£220 x 2 days (24 hrs)</td>
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<td>B1/12</td>
<td>Item</td>
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<td>sundry items which would be</td>
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<td>bill of quantities—duck bodies</td>
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<td>Value of items B1/2-9 inclusive</td>
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|           |          |      | $57462 \times 15\% = 8619  \\
| 2         |          |      | $6772
| 3         |          |      | $11,936
| 4         |          |      | $515
| 5         |          |      | $13,480
| 6         |          |      | $16,156
| 7         |          |      | $7124
| 8         |          |      | $694
| 9         |          |      | $785
|           |          |      | $57462 \times 15\% = 8619  \\

Say $9000
## TENDER ANALYSIS SHEET

**CONTRACT** E.W.P.P 79  
**SHEET NO.** 13

<table>
<thead>
<tr>
<th>I Ref</th>
<th>Quantity</th>
<th>Unit</th>
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<th>Rate</th>
<th>A-Labour</th>
<th>B-Plant</th>
<th>C-Material</th>
<th>D-Durable</th>
<th>E-Durable</th>
<th>F-Expense</th>
<th>H-S</th>
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<tbody>
<tr>
<td>1.1</td>
<td>3550 M2</td>
<td></td>
<td>Formwork circular, to inner and outer surfaces</td>
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</table>

Make: Br in steel moulds  
Current price/ft) $197/m²  
× 1.3 = $191-16/m²  
$191-10/m² ÷ 102 uses (5 sets total)  
Maintenance 5% of last  

Fix & Strike  
1.2L @ $3-73 Tender  
1.2L @ $3-05 lub:6  

Change in Prelims  
Rub down concrete surfaces  
 Allow  

Sunday consumables, tools e.t.c.  

| 1.7-91 | 1.5-40 | 1.0-05 | 1.9-60 | 0.50 |

---

Note: The table details the analysis of a tender for contract E.W.P.P 79, showing the quantity, rate, and various costs associated with different aspects of the project.
<table>
<thead>
<tr>
<th>II Ref.</th>
<th>Quantity</th>
<th>Unit</th>
<th>DETAILS</th>
<th>Rate</th>
<th>A-Labour</th>
<th>B-Plant</th>
<th>C-Material</th>
<th>D-Durable</th>
<th>E-Durable</th>
<th>F-Expense</th>
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<tr>
<td>2/2</td>
<td>127</td>
<td>Mg</td>
<td>Reinforcement - High yield</td>
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<td>Say: As B1/5</td>
<td>3.2817</td>
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<td>0.257.05</td>
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Note: The table continues with more entries, but they are not fully visible in the image.
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<th>A-Labour</th>
<th>B-Plant</th>
<th>C-Material</th>
<th>D-Durable</th>
<th>E-Durable</th>
<th>F-Expense</th>
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<td>82/3</td>
<td>1265 M3</td>
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<td>Reinforced concrete - 25 N</td>
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<td>OPC 0.360 T @ ≤ 34-36 Nett + 2½% waste</td>
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<td>12.68</td>
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<td>Frams 0.674 T @ ≤ 4.05 Nett + 2½% waste</td>
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<td>2.80</td>
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<td>Coarse 1.160 T @ ≤ 4.15 Nett + 2½% waste</td>
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<td>4.93</td>
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<td>Mix &amp; Transport : allow 25-20</td>
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<td>.39</td>
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<td>Place 0.8 hr @ ≤ 3.05</td>
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<td>1.47 .59</td>
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</table>
**Bill Ref.** | **Quantity** | **Unit** | **Details** | **Rate** | **A Labour** | **B Plant** | **C Material** | **D Durable** | **E Durable** | **F Expense** | **H S**
--- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | ---
22/4 | Item | | Prestressing | | | | | | | | |
<p>| | | | Materials 0.7” 18mm Dyform | | | | | | | | |
| | | | 105 Tendons each of 7 strands = 735 x | | | | | | | | |
| | | | 60.5 m Long (57.5 Nett + (2 x 15 m)) | | | | | | | | |
| | | | = 4.4468 m avg @ £0.945 Nett + 2% | | | | | | | | |
| | | | Sheathing | | | | | | | | |
| | | | 105 x 60.5 = 6352.5 m @ £0.94 Nett + 10% | | | | | | | | |
| | | | Anchorage 210 No @ 47.60 m Nett + 12% incl. wedges etc. | | | | | | | | |
| | | | Lab: Sheathing 6353 m ÷ 10 m/lt. @ 3.73 | | | | | | | | |
| | | | Threading cables &amp; fixing anchor | | | | | | | | |
| | | | 105 No x 10 hrs @ 3.73 | | | | | | | | |
| | | | Tensioning | | | | | | | | |
| | | | 105 No x 15 hrs @ 3.73 | | | | | | | | |
| | | | Hire of jacks Allow | | | | | | | | |
| | | | Grout 6353 m x π x 0.0375 | | | | | | | | |
| | | | 28 m³ | | | | | | | | |
| | | | @ 9150 1000 3000 200 | | | | | | | | |
| | | | £2562 £280 £840 | | | | | | | | |
| | | | Total | £77787 | £14724 | £2280 | £20727 | | | | | |</p>
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<td>Take delivery of end units and place in position before stressing</td>
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<td>Cranes in Prelim</td>
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<td>Lab attend</td>
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EDINBURGH-SCOPA-LAING

LAING

5TH YEAR

WAVE ENERGY

REPORT

APPENDIX 3

ALTERNATIVE BEAKS
MATT: PREMACHINED "EKKI" HARDWOOD. (33m³/DUCK)

PROPOSED WOODEN BEAK.

APPENDIX 3.

FOR STEEL BEAK SEE DRG NO 2022.

JOHN LAING LTD.
EDINBURGH-SCOPA-LAING

LAING

5TH YEAR

WAVE ENERGY

REPORT

APPENDIX 4

LISTS OF DRAWINGS
# List of Drawings

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EDINBURGH-SCOPA-LAING

LAING

5TH YEAR

WAVE ENERGY

REPORT

APPENDIX 5

CONTRIBUTING MANUFACTURERS WITH THEIR COMMENTS
APPENDIX 5
LIST OF COMPANIES PARTICIPATING

We gratefully acknowledge the interest and support of the following organisations, among others:-

Avon Rubber Company Limited
B.P. Oil Limited
Bestobell Seals Limited
Henry Berry Limited
Bradford Cylinders Limited
British Ropes Limited
British Steel Corporation
British Shipbuilders
Brown Brothers Limited
CERG Research Limited
Glacier Metal Company Limited
W. H. Gore Limited
Harland & Wolff Limited
Inco Europe Limited
F. H. Lloyd Limited
Markham & Company Limited
McTaggart Scott Limited
William Mallinson & Sons Limited
Monro & Miller Limited
Pirelli Cables Limited
Plastic Coatings Limited
Plastic Coating Research Co. Limited
RFD Inflatables Limited
Reinforced Plastic Structures Limited
Redpath Engineering Limited
Sacol Limited
Staffa (Chamberlain Industries Ltd.)
Sauer UK Limited
SUNTER/TTM Limited
Towler Hydraulics Limited
Whesoe Limited
Dunlop Limited

Melksham
London S.W.1
Slough
Leeds
Bradford
Doncaster
Glasgow
Newcastle
Edinburgh
Leatherhead
Alperton
Dunfermline
Belfast
London
Wednesbury
Chesterfield
Edinburgh
Hackney
Edinburgh
Eastleigh
Guildford
Camberley
Godalming
Lancing
Glengarnock
Totton
London
Lincoln
Rugby
London
Rodley
Darlington
Birmingham
Dear Peter,

Following our various discussions, I would like to confirm that I consider the design of the bearing pads described in my notes of the 19th November is feasible.

We would expect the cost of the bearing pads to be in the region of £500 to £1000 each. Both the diaphragm and the inflatable seal would be composed of rubber re-inforced with nylon cords.

Yours sincerely,

D.M. Turner,
Director, New Projects
Dear Sirs,

We apologise for the delay in sending you a written reply following our meeting at your offices on 16th October. However, we hope that the verbal comments passed to you in the interim provided sufficient information for your discussions at Mill Hill.

We will confirm that information in this letter but would like to add a cautionary note at this stage. Whilst we are confident that we can meet the two separate lubrication requirements, the suggestions put forward at this stage are first estimates as it were. We would anticipate that some slight modifications may be necessary as and when experience is gained under operational conditions. Bearing that point in mind we put forward the following:-

A. Technical Considerations

1. Hydraulic Fluid for Sealed Pod (for 25 year life)

For this application we suggest a special narrow cut low viscosity mineral oil, possibly containing an anti-wear or EP additive. The material has a boiling range typically from 260 - 335°C and a viscosity of approximately 11 cSt at 80°C (this we recall is somewhat higher than your indicated figure); however, the vapour pressure of this product is reasonably low - 3 x 10^{-3} mm Hg at 8°C - and with a system pressure of 5 mm Hg the oil should make no significant contribution to the pressure.

With regard to lubrication performance our literature searches have so far turned up little on the effects of inert (non-oxidising) atmospheres on anti-wear and extreme pressure properties of lubricants. Such work that has been done suggests that wear rates are less under nitrogen than under air; however, at higher loads friction becomes correspondingly higher, leading to seizure earlier than under aerobic conditions; hence the possible need for additive protection.
There appears to have been no work done on the minimum oxygen concentration required to overcome these effects or on whether carbon dioxide atmospheres are beneficial or harmful when compared with nitrogen or helium. However, the expertise is available within the Company should a programme of work be required to look into this aspect of the lubricant.

2. Lubricant for Linkages

We suggest that a conventional mineral oil based hydraulic fluid having good resistance to deposit formation in the presence of moisture should be a suitable starting point for this requirement. However, in view of the hostile sea water environment it is probable that enhanced corrosion inhibition will be required, and this could be accommodated without any problem.

B. Commercial Considerations

Accepting that development work on both products is likely and that some modifications will need to be made to our initial recommendations, the prices we have indicated below are best estimates. This information should enable you to complete your studies with a reasonable degree of accuracy.

Fluid (1) for the Sealed Pod - 38 PPL
Fluid (2) for the Linkages - 36 PPL

We wish you every success in your forthcoming presentation and look forward to hearing from you again in due course.

Yours faithfully,

for BP OIL LIMITED

Dr. R. Cecil
Mr. S. H. Salter,
University of Edinburgh,
Kings Buildings,
Mayfield Road,
Edinburgh, EH9 3JL

Dear Mr. Salter,

Hydraulic Seals - Wave Power Project

Having discussed the above subject at some length with all the appropriate people within the Company, our position can be defined as follows:-

Obviously, the present state of the art in sealing technology will not give you anywhere near the life which you desire. As you stated at the meeting in September, what is really needed is a device which acts not only as a controlled leakage component but also as a member which will temporarily generate a truly circular form in the inner sleeve of the cylinder so that no contact occurs between this sleeve and the piston.

We feel that elastomeric materials are unlikely to perform such duties satisfactorily and any move out of our present range of expertise is not considered to be in the best interests of the Company.

Having said this however, we shall be only too willing to assist in any work in which the use of elastomeric materials, within our expertise, is considered the best approach.

Yours sincerely,

H. Course,
Development Manager

Bestobell—an international group
Ref PH/LC/349

29 August 1979

The University of Edinburgh
School of Engineering Science
King's Buildings
Mayfield Road
EDINBURGH EH9 3JL

TERMS - Net and to be agreed, depending upon quantity ordered.

Dear Sirs

WAVE POWER PROJECT

We thank you for your enquiry of 21 June, calling for hydraulic cylinders and we have pleasure in submitting the following Specification and Tender:

One Cylinder Unit having the following characteristics:

Piston - 1,100/400 mm diameter x 2,000 mm stroke.

Working Pressure - 175 bar

Power - 1,650 tons

The construction of the cylinder would be as follows:

Cylinder of EN8 forged steel with fabricated mounting feet. The cylinder would be bored from the solid and honed. Lifting lugs would be welded to the cylinder.

Rear end cover of forged steel, having a machined spigot and drilled and fitted with retaining high tensile screws.

Front stuffing box of forged steel bored and fitted with a gunmetal neck bush and a gunmetal lined steel gland complete with twinset packings and also a wiper gland and packings. The stuffing box will be retained with high tensile screws.
Piston Rod manufactured from stainless steel type FV520B, polished on the working surface and screwed on the exposed end. The piston end will be machined to accept a forged steel piston head which will be complete with chevron packings and gunmetal packing rings. The head would be retained with a forged steel nut.

Piston Rod Eye - We include for a cast steel piston rod eye which will be screwed and mounted on the piston rod end. The eye will be bored and fitted with a gunmetal bush 650 mm diameter.

Our offer does not include for cylinder mounting bolts.

Hydraulic tests - The cylinder would be hydraulically tested in our works before despatch.

Painting - The cylinder would be specially painted to suit the working conditions.

BUDGET PRICE - £127,000 net.

If a multiple number of cylinders are ordered together, this price could be reduced by £3,000.

As an alternative, we offer a similar cylinder unit having the following characteristics:

Piston - 1,100/500 mm diameter x 2,000 mm stroke.

Working Pressure - 275 bar

Power - 2,650 tons

The construction of the cylinder would be as previously described.

PRICE - £172,000 each net.

If multiple cylinders are ordered together, this price can be reduced by £3,000.

We now offer:

One cylinder unit having the following characteristics:

Piston - 580/250 mm diameter by 2,000 mm stroke.

Working Pressure - 175 bar

Power - 490 tons

The construction of the cylinder would be as follows:
Cylinder of forged steel EN8 quality bored from the solid and honed.

Rear end cover of cast steel having a machined spigot for mounting purposes and retained with high tensile steel screws.

The cover would have an eye piece which would be bored and fitted with a gunmetal bush 350 mm diameter.

Stuffing box in the front of the cylinder would be mounted a forged steel stuffing box retained with high tensile screws. The stuffing box would be bored and fitted with a gunmetal neck bush and a gunmetal lined steel gland complete with twinset packings. There would also be a wiper gland and packing.

Piston Rod of stainless steel of FV520B quality smooth machined on the working surface and screw cut on the exposed end. The piston end would be complete with a forged steel piston head fitted with chevron packing and gunmetal packing ring and the head would be retained with a forged steel locking nut.

Lifting lugs would be welded to the cylinder unit.

Piston Rod Eye - We include for a cast steel piston rod eye screwed and mounted on the piston rod end and the eye bored and fitted with a gunmetal bush 350 mm diameter.

Hydraulic tests - The cylinder would be hydraulically tested before despatch.

Painting - The outside of the cylinder would be specially painted to suit the working conditions.

PRICE - £57,000 each net.

If multiple cylinders are ordered at the same time, the price can be reduced by £2,000 each.

Finally, we offer one cylinder unit having the following characteristics:

Piston - 580/320 mm diameter x 2,000 mm stroke.

Working Pressure - 275 bar

Power - 785 tons

The cylinder construction would be as previously described.

PRICE - £73,000 each net.

If multiple cylinders are ordered at the same time, this price could be reduced by £2,000 each.
All the above prices are budget prices and exclude VAT and would be subject to Escalation.

**DELIVERY** - Scotland

**DESPATCH** - This would depend upon the quantity ordered but the first cylinders would require 12 months.

We enclose a copy of our drawing 79074T illustrating the cylinders, herewith.

We trust our offer proves of interest and look forward to hearing further from you in due course.

Yours faithfully

HENRY BERRY & CO LTD

Managing Director
Dear Sirs,

Confirming our telex dated: 31.10.1979

We thank you for the above enquiry and have pleasure in quoting as follows:

One off - Bradford Hydraulic Cylinder (NODDING DUCKS), 700mm bore x 240mm diameter x 1000mm stroke of the double-acting class. Non-cushioned with through piston rod - one end of rod to be plain with rod extended. Suitable for a working pressure of 3000 lbs. psi. As shown on our enclosed print of General Arrangement Drawing BC 18468 - Your Drawing Ref. 100102/MA1017.

Price: £9,617.00.

Extra for self-aligning rod clevis

Price: £2,000.00.

Extra for two way trunnion.

Price: £2,500.00.

Price for quantities of 20 off or more would be subject to a discount of 15%.

Validity: 30 days. Prices subject to BEAMA CPA Escalation from expiry of validity.

THIS QUOTATION DOES NOT INCLUDE V.A.T.

PLEASE NOTE: To cover the cost of documentation a surcharge of £5.00 will be added to orders of less than £50.00 in value.

DELIVERY: 16/20 wkg. weeks from receipt of your order subject to confirmation at that time.

TEILNS: 30 days nett from date of despatch (packing and carriage charged extra).

We trust that this offer is acceptable and look forward to receiving your order instructions in the near future.

This cylinder carries a 25 year guarantee for workmanship and material.

Yours faithfully,

for and on behalf of

BRADFORD CYLINDERS LTD

P Simms
Director & Works Manager

Sales Office Manager

A Member of the Thorn Group

THORN

Regd. in England No.1230777

CONDITIONS OF SALE OVERLEAF
For the attention of F.S. Nundy, Esq.,

Dear Sir,

Further to our meeting in Mill Hill we have now had an opportunity to examine the loading conditions for the Salter Duck and feel that we are in a position to express an opinion on the feasibility of the mooring system.

The figures you have supplied to us should not present any problems as far as tension fatigue is concerned, even with an increase of 10% for the dynamic condition. However, as we discussed at our meeting, the area which is most likely to cause concern is that of bend fatigue. If, as you suggest, potential bend fatigue problems can be designed out of the system, we agree in principle that a fatigue life of 25 years can be achieved for the mooring lines, using conventional rope-making materials and manufacturing techniques. We are also confident that a wire rope or strand suitably protected will have a corrosive life of 25 years.

We realise that this project is still in a relatively early stage but feel that close co-operation between our two organisations is essential in order to recognise any potential problems in the mooring system. Meanwhile, if we can be of any further assistance please do not hesitate to contact us.

Yours faithfully,

K. T. RONSON
SENIOR ENGINEER.

cc. Mr. M. Bullas
University of Edinburgh
Kings Buildings, Mayfield Rd.,
Edinburgh EH9 3JL.

cc. V.J. Harris, Esq.,
c.f.i. H. T. Plant, Esq.,
D. M. Sharp, Esq.,
Dear Mr. Freer,

This letter is to record my thanks to you and your colleagues for the very interesting visit I had on Tuesday.

I have started seeking additional information as requested and have asked colleagues to get in touch with you directly concerning the flywheel axis, bearings and large rings.

As we thought the size of the spine is well outwith Stanton and Staveley's capability and my contact there agreed it would need to be made by segmental concrete construction. A company he suggested could tackle it was Kinnear Moodie of Peterborough, part of the Tarmac Group. Incidentally my impression was that concrete was chosen for the spine before the gyro solution was evolved, which does not impose the same loadings on the spine. Also the spine lengths are shorter (only 2 ducks per spine length) which imposes less strain on the individual spine lengths. Perhaps when my colleagues in Redpath Engineering speak to you, you could seek their opinion as to whether steel could again be considered.

Concerning corrosion, would it be possible to use some of the electricity generated to cathodically protect the structure? I do not think it would lead to a great loss in efficiency. You asked if there was any method of bulk plastic spraying. I have just found out from my colleagues in Redpath Engineering that a firm in Dundee can provide this service and they will forward the relevant literature.

With regard to more conventional systems it is claimed by a company called Metallisation Limited that 175 microns sprayed zinc plus a two pack primer plus a two pack polyurethane top coat will give a life to first maintenance of 25 years or a total life of 40-50 years. To what degree they can substantiate their claims I do not know. We are investigating this and whether a sprayed metal coating is compatible with impressed current cathodic protection.
The surface area (24 m by 24 m) of the peak indicated to me, at a thickness of say 12 mm would weigh about 55 tonnes. This daily requirement is well in excess of our current capability to produce clad plate (it is about equal to our weekly capacity) and we would have to give serious consideration to enhancing our facilities if your system does go into the mass production phase. There should be no problems providing the quantities required for your prototypes given a certain amount of notice. I cannot yet give precise prices for cupronickel clad plate as we have not yet finally built up our production route but if you base any calculations on about £1,750 per tonne you should not be too far out.

Cupronickel cladding is the only method I can suggest that will give resistance to fouling over a long period. Antifouling paints typically require renewing every 5-7 years.

Since you will have the facility to ballast up and down can you not trim the ducks to compensate for marine growth weight or is the weight not the problem.

I will revert at a later date concerning material for the flywheels.

If the above gives rise to any questions please do not hesitate to get in touch with me.

Yours sincerely

R DICKSON

Prices quoted: Plank 854360
Grade 43A

£ 184/ton
39 Thickness/Width Excl
4 Length Excl

£ 227

2/10/79

AM02/M1
Dear Mr. Williams,

EDINBURGH WAVE ENERGY PROJECT

Firstly let me apologise for the long delay in answering your letter of the 12th September, there have been many reasons for this which I do not propose to bore you with.

I have examined the drawings you provided and so far as I can judge, while it would be a difficult and somewhat complicated job to manufacture it should be possible to do so.

In reviewing the possibility of Brown Bros. being involved in the manufacture of this work I have to advise you that it would not be possible to provide facilities and space for either the prototypes or the type of production numbers you envisage. These kinds of numbers would almost require a factory of some size to be doing nothing else but this work.

I will, however, forward the drawings to the Vickers Organisation to see whether or no they would be interested in becoming involved.

I am sorry that we find it impossible to be part of this project but we have to be certain that we do not prejudice the manufacture of our own products which usually keep our factory very full indeed.

Yours sincerely,

A. POTTINGER
PRODUCTION DIRECTOR
12 October '79

Mr R Freer
University of Edinburgh
School of Engineering Science
King's Buildings
Mayfield Road
Edinburgh EH9 3JL

Dear Mr Freer

EDINBURGH UNIVERSITY WAVE POWER DEVICE

Thank you for your letter of 26th September, concerning the possible use of polyurethane tyres as roller bearings between the Duck and the spine. Dunlop make a range of tyres of approximately 0.4m diameter, and of these the maximum load carrying ability is given by a 16 x 5 - 10½ "Duthane" tyre. The quoted maximum load is 18000 lb, but Mr E Lees (R & D Manager, GRG Division, Wrexham) feels that in the anticipated service conditions, this should be reduced to 9250 lb (4.2 kg). This assumes continuous running at maximum load, at a speed approaching 10 mph. The maximum load is effectively that required to produce a radial deflection (10% or 15%) which is known to cause overheating of the tyres. Hence continuous running at high loads is likely to cause problems, although cooling is likely to be improved in sea water.

Remembering that only about one quarter of the tyres carry load at any instant, the total number of tyres required can be estimated as follows:

- Length of Wave Power Device = 30 m
- Load per unit length = 20 ton/m
- *". Total load = 600 ton
- Maximum load per tyre = 9250 lb
- *". Number of tyre carrying load = 145
- Total number of tyres = 580

or 12 rings of 48 tyres each.

If the service conditions are significantly less severe than has been assumed, the number of tyres could be reduced accordingly.

The list price of these tyres (1978 catalogue) is £159.84 each, but for the quantities envisaged, this would be reduced to about £90, making a total cost of £52 200 or £1740/m.

Cont/...
Although I understand the advantages of using tyres in this "ball-race" fashion, I feel that I should not close without reiterating some of the disadvantages:

(i) Very high precision is required in fabricating the bearing surfaces, otherwise uneven load distribution would lead to early tyre failure.

(ii) Slip appears to be involved in the rolling of rings of tyres, linked together at their axles, between cylindrical surfaces. This would lead to drag (reducing the efficiency of the device), and to tyre wear, which would be likely to be unevenly distributed around the periphery. (Slip is not involved if each tyre is free to rotate about an axis suitably displaced towards the centre of the Duck).

(iii) In the event of significant abrasion, it might be necessary to replace all the tyres in the ring, otherwise the new (larger) tyres could be overloaded.

(iv) To carry load the tyres need to be deflected. Support to the spine would probably be better if tyres made contact all round the periphery. This implies that they need to be pre-loaded radially inwards, which is difficult with the roller-bearing system.

(v) A roller-bearing device implies two deflections per revolution of the tyre, doubling the heat build-up.

All these disadvantages can be obviated by mounting the tyres on simple, spring loaded suspension systems.

Yours sincerely
For DUNLOP LIMITED

M H Walters
Physical Research Department

Copy - Mr E Lees
Ingeco Laing Ltd.,
Farrow House,
Colindeep Lane,
LONDON NW9 6HE

For the attention of Mr. P.B. Williams

Dear Sirs,

Wave Power Project

Following our discussions over the last few months, and our visit to Edinburgh University on 5th September, 1979, we have pleasure in confirming our willingness to consider supplying 3300V, 500A, 99-way electrical connector assemblies capable of being mated and sealed underwater. They would be generally similar to our drawing WP.6, a print of which is enclosed herewith, but with such changes as might have been deemed advantageous by the time the need for their production arrived.

Our first, interim, estimate of the probable costs of the various phases of the project have already been conveyed to you by telex, but will be repeated here:

- Design and development: circa £50,000
- Tooling costs: circa £15,000
- Unit cost, per capita: circa £12,000

All the above are quoted in pounds Sterling. When we have received more definitive quotations from our suppliers of bought-out components, notably castings, we will be in a position to give a more precise unit price; it is expected that the final figure will be somewhat lower than the figure quoted above.

It will naturally be appreciated by your good selves that our participation in this activity would be dependent on the agreement of mutually acceptable terms of Contract between the various parties involved.

We trust this letter will be of assistance to you in the furtherance of your project studies of the overall wave power concept.

Yours faithfully

FLIGHT REFUELLING LIMITED

J.E. Medgett

Engineering Manager
Military Systems Division

Enc.
Dear Sirs

WODDING DUCK WAVE ENERGY PROJECT

Further to our telex of the 1st November 79, we confirm for guidance purposes only an estimated total cost of £150 000 to £200 000 for 4 bearing units. For the present one unit is assumed to comprise the 5 items shown on the righthand side of our Sketch SK/NHN/8041/1, materials as yet unspecified.

Our comments with regard to the main journal bearings on the fly wheel assembly, as discussed in our offices on the 14th November 79, are as follows:

- Our preliminary estimates indicate that for full hydrodynamic action bearings of at least 500 mm diameter x 500 mm long must be used. At this size the loading is higher than normal industrial practice but provided that very high standards of alignment, filtration and shaft surface finish are maintained, the bearings should be satisfactory. Calculations using an ISO VG 3 oil suggest that losses in the order of 60 kW per bearing could be expected.

- By introducing some high pressure oil into the bearing, but not making it a full hydrostatic bearing, sufficient combined hydrodynamic/hydrostatic action could probably be obtained on a reduced bearing diameter of 300/350 mm. Initial estimates give a shear loss of about 22 kW per bearing but we do not have an estimate of the hydrostatic pumping losses. Please let us know if you wish us to take this idea further.

In general it may well be that present bearing technology is inadequate for this project. However, we are certainly interested in co-operating with you in finding solutions to the various problems and are of the belief that we would be able to develop a bearing system with the necessary life and reliability for the application.

Yours faithfully,

[Signature]

SMITH - COMMERCIAL MANAGER - MARINE & HEAVY ENGINEERING SALES

Registration London 166 096 Registered Office 368 Ealing Road Alperton Wembley Middlesex England
FOR THE ATTENTION OF MR P B WILLIAMS

Ingeco Laing Limited
Farrow House
Colindeep Lane
Collindale
London

Dear Mr Williams

Following my recent visit to discuss castings for the "Edinburgh Duck" project, I have pleasure in confirming that we foresee no difficulty in producing to your requirements.

I would also like to take this opportunity of inviting you, and any of your colleagues who may be interested, to our Works at a convenient time to discuss in greater detail the areas, and expense to which we are able to help you.

Mr D C Lloyd and myself would like to take up your invitation to visit the test tank etc. at Edinburgh, and would welcome your suggestions as to convenient dates.

Kindest regards,

Yours sincerely

JS Smith

A subsidiary of F. H. Lloyd Holdines Limited
Dear Mr. Williams,

Re: Wave Power Project.

Thank you for a most interesting explanation of your Wave Power Project last Friday, and for the relevant information on the possible use of timber for cladding the beak of the duck module.

As suggested, the marine constructional timber "EKKI" (Lophira Alata), which is grown in West Africa, would seem to be the timber for the purpose for which it is required, based on the following brief reasons:-

"EKKI".

(1) Is a hardwood which has been used extensively for marine purposes for over 30 years, for jetty fenders, dolphins, dock gates, sea and river protection work, also many other uses.

(2) "Ekki" has constant physical properties with high working stress figures. See attached details.

(3) The timber is resistant to abrasion, is very tough, reputed to be the most durable timber from West Africa, performs exceedingly well in polluted waters without treatment - when subjected to wetting and drying (as in your case).

(4) "Ekki" is particularly resistant to attack by marine borers, such as the Teredo borer and Limnoria Gribble. This could be an important aspect, as the Limnoria Gribble is generally active around the Coast of Britain.

(5) "Ekki" is commercially available and is relatively cheap, as it is not used for veneers, plywood and joinery purposes due to its density.

As far as my Company is concerned, we can offer "Ekki" sawn to your sizes from our associated saw mill in Holland, where some three to four thousand m³ of "Ekki" logs are always held in stock.

The conversion of "Ekki" logs to sawn sizes is the sawmill's primary function. In addition, machining and assembly facilities are available for the production of complete units, currently Lock/Dock Gates, as an example.

18th October, 1979.
(6) As a detailed point of interest, we have supplied "Ekki" as fenders for dolphins which are situated off the entrance to the Leith Docks in Scotland.

A dolphin was rammed by a ship recently, replacement "Ekki" is being supplied this month, so the complete dolphin should be available for inspection in Leith Docks early November, 1979.

Would you like to see the dolphin?

(7) Since our meeting, I have checked the current prices against the sawn sizes, as under, and based on complete trailer loads of 15m³ delivered to the Oban area, an overall guide price could be taken as £350.00 m³. If a Coaster was organised then the costs should be slightly cheaper. I will revert to this as soon as prices are available.

The price is based on :- Complete loads, an Exchange Rate of 4.28 Dutch Guilders to the £ Sterling, on the current freight/handling rates, with any variation in these rates, rise or fall, at the time of delivery, or invoicing as appropriate, will be to buyer's account.

Machining, such as drilling of holes, would be a variation on the guide price.

Sizes per 'duck'

Decking - 150, 200, 250mm. width x 38mm. thickness in length 4 - 7m. mults. of 1m.

Longitudinals - (Sawn shape) - 280 x 280mm. in 5 - 7 m. length.

Framing - 300 x 100mm. x 7m. length.

Say 33m³ per duck. @ £350/m³ (7) = £11,500 /duck.

We would be pleased to undertake the supply of "Ekki" required for a prototype duck. The lead time from final sizes could be taken as 6 weeks or so.

Syncrolift. - The designers of the Syncrolift System are :-

Pearlson Engineering Company Inc.,
17 Devonshire Street,
Telephone : 323-2855

Mr.H.C.Mackenzie Wilson is the Managing Director.

I hope I have covered the information which you require and now look forward to a further discussion in Edinburgh. In the meantime, if you require any further information do please let me know.

Yours sincerely,
for and on behalf of
WILLIAM MALLINSON & SONS (SALES) LTD.

L.W.Pollard.
## TECHNICAL INFORMATION.

**"Ekki" - Lophira Alata - Physical and Mechanical Properties.**

**Ultimate figures.**

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density - Green</td>
<td>1314 Kg/m³</td>
</tr>
<tr>
<td>Specific Gravity</td>
<td>0.88</td>
</tr>
<tr>
<td>Modulus of rupture</td>
<td>123 N/mm²</td>
</tr>
<tr>
<td>Modulus of Elasticity</td>
<td>13900 N/mm²</td>
</tr>
<tr>
<td>Resistance to suddenly applied loads</td>
<td>1.40 m</td>
</tr>
<tr>
<td>Compression parallel to grain</td>
<td>68.4 N/mm²</td>
</tr>
<tr>
<td>Resistance to indentation</td>
<td>12850 N</td>
</tr>
<tr>
<td>Maximum Shearing Strength parallel to grain</td>
<td>16 N/mm²</td>
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</tbody>
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**WORKING STRESSES at 30% Moisture Content in Kg/cm²**

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
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<td>Bending</td>
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<tr>
<td>Tension</td>
<td>230</td>
</tr>
<tr>
<td>Compression parallel to grain</td>
<td>220</td>
</tr>
<tr>
<td>Compression parallel to grain at right angles</td>
<td>80</td>
</tr>
<tr>
<td>Shear</td>
<td>20</td>
</tr>
<tr>
<td>Modulus of elasticity</td>
<td>170,000</td>
</tr>
</tbody>
</table>
Ingeco Laing Limited
Farrow House
Colindeep Lane
London
NW9 6HE

For the attention of Mr. P.B. Williams
Engineering Consultant

Dear Peter

Tidal Energy - Nodding Duck Project

Firstly, may I say how pleasant it was to meet up with you again, and particularly with an involvement on such an interesting project. Our discussions so far have been quite refreshing in that we have been introduced to a new technology, novel in conception and challenging in size.

We have been asked to look at the manufacturing feasibility of the flywheel units, and to indicate the extent of our interest in a manufacturing involvement. I have discussed the scheme with a number of our people, and the general thoughts outlined below are based on an understanding that basic parameters are as follows:

1) Consideration to be given to flywheel units only, excluding hydraulic motors, and additionally also to manufacture of the equipment containment vessel.

2) Overall project to be considered on the basis of a 2000 MW string of ducks at 75 KW/M wave front generation requiring 3300 ducks.

3) Each duck will carry four flywheel units with laminated flywheels of 3.66m diameter, each flywheel weighing approximately 17 tonnes.

4) Total weight per flywheel assembly 50 tonnes. Four assemblies required per duck.

5) Containment vessel to be approximately 5.1m diameter of 25 mm plate, approximate weight per vessel 100 tonnes.

6) Time scale for construction of a 2000 MW unit to be comparable with conventional power station build times.
Ingeco Laing Limited
London
22nd August, 1979

Our comments and views are as follows:

1. We would be very interested to participate both in the prototype and development, and also full scale production stages.

2. The sheer scale of production of 3300 ducks involving 13200 flywheel assemblies (and we are considering only the flywheel assemblies) requires a volume production approach with which heavy engineering companies are unfamiliar (equally, production engineers versed in volume production are unlikely to have any heavy engineering experience). An initial appraisal, based on our own workforce of 750 total and works equipment, and assuming that 50% of our capacity was devoted to duck flywheels, and also assuming that our normal jobbing approach was taken the production engineering, showed that we could produce approximately 40 flywheels per year, or 3% of requirements based on a 10 year production programme.

This could probably be increased by 50 to 100% with a properly considered production system, supplemented with provision of all requisite machine tools, production jigs and fixtures.

Accordingly, the amount of heavy capacity which would be needed to be devoted to the project is not inconsiderable, but should be available among heavy plant builders, boilermakers, shipyards, etc. throughout the country, and could have something of a rejuvenating effect on an industry which has been passing through dispiriting times.

3. The production of containment vessels are considered to be practicable only at a waterside works. Transportation of 5.1m diameter vessels at 100 tons each from inland works to an assembly yard in Scotland does not seem feasible in any quantity. Additionally, the sheer floor space required can best be found at an assembly yard - which can probably best be an offshore rig building yard or shipyard in Scotland suitably converted. (The John Brown offshore yard at Rothersey Dock comes to mind as being typical.)

One suggestion was that the containment vessel need not be so strongly constructed as proposed - it could be a thin skinned, or even fibreglass vessel cast into the concrete duck structure.

Overall, it was felt that the most practical arrangement, taking into account transport, existing and mothballed U.K. capacity and availability of labour, would be to build the vessels and assemble the internals at a waterside yard established specially, or converted to suit the duck project. Specialised parts, such as flywheel assemblies, pump, motors, etc. to be supplied to the yard by specialist manufacturers.
3. Vessel manufacture and assembly on the basis of 330 units per year would require a yard labour force of around 5000 and possibly more.

4. Supply of materials would need positive commitments on the part of sub suppliers such as B.S.C. (for the flywheel laminates) forges for the shafts and foundries for the mounting castings.

5. To determine an overall price per unit would require a careful assessment of jigs and special machinery required, which in turn would require a frozen design to work to. However, the notional figure of £5,000.00/tonne at today's level should cover the flywheel assemblies. It is doubtful if this would be adequate for the pump and motors, however. This again would depend on production engineering.

Summarising, we would like to confirm our interest as suppliers in the Nodding Duck Project. The designs we have examined are in an early project stage, but are in general feasible from a manufacturing aspect. To achieve reasonable prices and deliveries, careful production engineering analysis will be necessary, together with a substantial investment in machine tools and jigs.

We would be happy to be involved in both prototype and production stages.

Please keep us in touch with developments.

Yours sincerely

[Signature]

K. Wort
(Director Markham & Co Ltd)
Dear Mr. Salter,

We found the Wave Energy demonstration, which we attended today, to be extremely interesting and thank you for giving us the opportunity to attend.

Since the meeting in our offices, we have given considerable thought to arriving at a budget price for the manufacture of suitable expansion joints for your project. As you obviously appreciate, this is a difficult exercise to carry out and the figures we give must be considered very approximate.

We have assumed that the expansion joints will be manufactured on the same site as the "ducks" and that suitably covered workshop facilities will be made available to us free of charge. We have made no provision for mechanical handling, power, lighting etc.

By extrapolating our existing manufacturing times, we are of the opinion that by utilising one longitudinal seam welding machine and one corrugation roll forming machine and working on a 3 shift basis, 345 expansion joints per year could be manufactured, a year being 5 full 24 hour shifts per week and 46 working weeks. Obviously, there is room for an increase in this work rate by working weekends but we have not taken this into consideration at this stage.

On this basis, we visualise that the capital cost would be between £150,000 and £200,000 and that each element would cost in the region of £8,000 to £10,000. If you consider this set-up to be one production line then the capital cost would simply be multiplied by the number of production lines necessary to reach the desired annual figure i.e., if the total requirement is 1,000 per year then the capital cost should be increased by a factor of 3. The price per element will, of course, remain as estimated.

We /
We trust this information meets your requirements. If you need any further details, please telephone the undersigned.

We apologise that it was necessary for Mr. Munro, Mr. Sinclair and the writer to leave the demonstration before the end. However, we had a prior engagement which made this necessary.

We look forward to hearing from you.

Yours sincerely,

R.F. Ewen
Technical Manager
1. On the instigation of Mr. Williams of Laings, a Meeting was arranged with Mr. J.E. Mudgett at Flight Refuelling on the 9th August to discuss the possible involvement of F.R. in the Salter Duck Project.

2. F.R. are a Precision Engineering Company engaged in the manufacture of military and civil aircraft refuelling systems and both towed and free flight targets. They also have a contract from the C.E.G.B. for the manufacture of a remote visual inspection device for the graphite cores of nuclear reactors.

3. F.R. involvement in the Duck Project would be to design and develop a 75 pin 3.3kV plug and socket suitable for 500 A. This would be used to connect individual Duck modules to a transformer pod situated mid-way along the spine.

4. There is little doubt that F.R. have the necessary expertise to undertake the mechanical side of this work and Pirelli would prepared to advise on the electrical aspects.

5. I suggested that their Managing Director discuss their possible involvement with the ETSU Project Officer for the Duck so that he could gauge both the volume of work and its potential.

J. McConnell 12/3/73
Mr. Williams,
Ingeco Laing Limited,
Farrow House,
Colindeep Lane,
London, NW9 6HE.

Dear Mr. Williams,

Thank you for your hospitality on Monday, I found our meeting most stimulating.

I feel that there are several areas where we could be of assistance apart from the obvious one of corrosion proofing the pressure vessels. You will recall that we discussed the gaiters for the hydraulic rams, coatings for steel fasteners to avoid the use of stainless steel and coated rebar for the concrete construction.

I have taken the liberty of sending you publicity and a sample of both the PTFE/Cadmium plated stud bolt and the dip moulded P.V.C. Gaiter, and hope you will find these of interest.

I look forward to seeing you in Edinburgh.

Yours sincerely,

Ian Marcham
Marketing Manager
P. B. Williams Esq.,
Wave Energy Project,
Ingeco Lang Ltd.,
Farrow House,
Colinkeep Lane,

Dear Mr. Williams,

Further to our telephone conversation of yesterday, I enclose herewith a data sheet on the reformulated Vacsol.

As you will see the chemical properties of the coating remain unchanged, but the content of non-volatile solids has been increased by 35% and the liquid is now considerably easier to apply by brush and, with this method, single dry films of 0.005" can be achieved with ease.

I have also despatched per parcel post to the above address, an 8 oz. liquid sample of Vacsol for your evaluation.

Should you require any further information, please do not hesitate to contact us.

Yours sincerely,
THE PLASTIC COATING RESEARCH CO. LTD.

J. H. Blakely.
Wave Energy Project

We would like to thank you for your approach to us regarding the above, and your visit on 19th June 1979. We welcome every potential opportunity for involvement in new fields of activity.

In broad terms the problem which you drew to our attention was that of providing bending constraint between the individual 'ducks' constituting the wave energy extractors. In particular you showed us a scheme of ball and socket interlinking, with the suggestion that a yoke of fluid-inflated pressure bags might provide the features sought. If we understood these correctly, included were:-

(1) the greatest freedom for individual ducks to rotate,

(2) a basic moment characteristic with angular deflection of rapid initial rise followed by an almost constant level until the mechanical limit was reached,

(3) a bag internal pressure of order 100 p.s.i. for bag contact dimension of 15 ft. (full scale values),

(4) the potential for fluid transfer between bags as a means of imparting desired response or behaviour patterns.

As well as the ball and socket/pressure bag concept, you will recall that our discussions ranged widely over possible alternative approaches including spring loaded wheels, disc springs etc., and a possible alleviation of the requirements due to both sea irregularity and inertia effects.

In our opinion it is not inconceivable that pressure bags suitable for the project could be satisfactorily designed and developed within a five year timescale provided this effort was supported by sufficient funding. However, since the project would ultimately demand strength and ruggedness of a very high order to provide guaranteed function over the period envisaged, we would suggest that you should seek further advice from others already engaged in the manufacture of ships fendering and other similar heavy-duty products.
We also strongly believe that re-examination of the use of more orthodox mechanical solutions should be conducted before any pressure bag scheme is adopted in principle.

Should you decide to proceed with a pressure bag approach, we would be pleased to consider tendering for a development programme.

Yours sincerely,

D.V. Edwards
Technical Director.
Mr. Bullas,  
Edinburgh University -  
Wave Energy Project,  
Department of Mechanical Engineering,  
Kings Building,  
Mayfield Road,  
EDINBURGH 9.  

Dear Mr. Bullas,  

Sub Surface Spheres.

My apologies to you for the delay in writing, but I am now pleased to confirm the text of our conversation of the 6th November.

Given that the information available to me is extremely limited and that there was no time for close examination of the problems of production etc., I was able to offer you a price of £65.00 per metre square for a 12mm thick laminate, this price takes into consideration the fact that there may be large numbers of these items involved.

We spoke of the volumes and weights of foam, and for a 16 ft diameter sphere it was estimated that nearly 10,000 lbs of 45 lb per cu.ft. density foam would be required. The cost of this to me would be in the order of £5,500 - £6,000 (excluding forming labour).

I trust that this covers the matter to date and I look forward to hearing further from you.

Yours faithfully,  
REINFORCED PLASTIC STRUCTURES (LEWES) LTD.

D.C. MULLOR  
Works Director.
Ingeco Laing Limited,  
Farrow House, 
Colindeep Lane,  
LONDON. NW9 6HE

For the attention of MR. P.B. WILLIAMS - Consultant Engineer

Dear Sirs,

EDINBURGH WAVE ENERGY PROJECT

We thank you for your letter and Drawings showing the complete Spine Joint for the above Project.

Having looked over the Drawings we noted the similarity to the 5 M Wind Tunnel Sections which we fabricated for you in the past. We, therefore, see no reason why this project should not be feasible using your design and engineering and our fabrication and practical experience.

When further details are known it would be to our mutual advantage possibly to discuss such things as keeping material qualities and thicknesses constant with the object of minimising costs.

Our best Indicative Price at this moment in time is £1,000.00 per Tonne.

We hope the above is of use to you and, should you require further information, please do not hesitate to contact us.

Yours faithfully,

For REDPATH ENGINEERING LIMITED

An RDL Company trading as agent for Redpath Dorman Long Ltd, a subsidiary of British Steel Corporation Incorporated in England, Reg No 250183. Reg Office 53 Goldington Road, Bedford MK40 3LR
Dear Mr. Williams,

With reference to your telephone call concerning the "Nodding Duck" project, after due deliberation of our long term commitments we find that we are unable to consider manufacture of the pump.

Thank you for giving us the opportunity of being involved in this interesting project.

Yours sincerely,

P. J. Chapple
Technical Director
Dear Mr. Salter,

Further to the various discussions we have had regarding the possible use of 'Sintox' ceramics in your Wave Generators, we have now looked at the ramifications of manufacturing production quantities of components to your drawing A.74060 and SKD.12. Technically, there is no reason why we cannot manufacture the piston and cylinder sleeve, but, to cope with the proposed quantities, we would, of course, need to look closely at our available capacity. Since this is the kind of work with which we should like to be associated, and bearing in mind that it is likely to be a long-term project, we believe that any adjustments, which we would need to make, would not cause supply problems, that is, provided we are able to work closely with the eventual contractor.

As a guide, small quantity samples of both items are likely to cost around £70.00 each, and for large quantities, about £30.00 each for the cylinder and £20.00 each for the piston. These prices are Nett and Ex Works and are based on costs, using our best available method of manufacture.

We shall be interested to hear if these prices are in line with your own ideas, and also your up-dated views on whether the whole project is likely to "take off", bearing in mind the fact that the alternative methods of generating electricity are getting more expensive.

Yours sincerely,

SMITHS INDUSTRIES LIMITED

John Williams
Sales Manager
Ceramics Division

MANUFACTURERS OF 'HYLUMINA' AND 'SINTOX' INDUSTRIAL CERAMICS
Registered Office: Cricklewood, London NW2 6JN  Incorporated in England  No. 137013
2nd November, 1979

Ref: DDS/js/I50

Ingeco Laing Ltd.,
Farrow House,
Colindeep Lane,

For the attention of Mr. J. Ling, Construction Manager

Dear Sirs,

Transportation of Salter Duck Assemblies

Further to our recent meeting at your offices where we discussed various aspects of transport, assembly, installation and mooring of the Salter Ducks, we wish to confirm that the methods proposed are all feasible using equipment and techniques available today.

We have produced schematic sketches enclosed which indicate land and marine movement of duck 'beaks': skidding of the complete duck assemblies at the construction site and typical details of equipment, all owned and operated by Sunter-ITM, which would be ideally suited to the project.

Our activities in the offshore industries have involved many of the waterside fabrication yards throughout Northern Europe, which specialise in the manufacture of large steel structures. This enables us to include in this document a list of possible yards where fabrication of beaks, etc., could take place.

The following brief notes confirm the various methods discussed at our meeting:
1. **Transportation of Duck Beaks from Fabricator to Construction Site**

The duck beaks having dimensions as you advised would weigh approx. 500 tonnes each: indeed the techniques we would use could easily accept this weight increasing to, say, 1000 tones without detriment.

The fabricator would need a waterside facility capable of taking a 300' X 90' flat top barge. Modular hydraulic trailers would collect the duck beak from its fabrication position, travel to quay edge, and load onto the barge. The beak would be loaded onto permanent steel supports and the trailer offloaded. Seafastening would commence immediately. The barge would accept a total of six beaks which after seafastening would be towed by a single ocean going tug to the construction site.

For a yard with large tidal range, loading would be restricted to one per day: where tidal range was minimal or the yard situated in an impounded dock then two beaks could be loaded per day.

2. **Offloading at the Construction Site**

The construction site will be required to have a roll on/roll off quay suitable to accommodate the 300' X 90' barge. Modular trailer equipment, to satisfy the large number of beaks to be delivered, would be permanently based on the site and would offload the beaks from the barge moored afloat, end-on to the quay.

3. **Launching of Fully Assembled Ducks**

We enclose details of the "Macskid" skidding equipment which has the ability to move very heavy loads on steel runway beams using hydraulic gripper jacks as the motive force. The completed 3000 tonne ducks would be skidded from the final construction position either:

a. directly into the water if the site location allows this, or

b. onto a launch barge which would then be taken to the nearest deep water and the duck off-loaded.
4. **Mooring of Duck Fleets Offshore**

We have not yet completed our investigations into mooring systems and equipment types, however, we again confirm that the proposals tabled are entirely feasible.

Further investigation will indicate the most suitable arrangement of mooring lines. The greatest problem as we see it will be the positioning of the anchors, therefore we are considering a type of anchor widely used for the mooring of offshore structures where sandy sea bed conditions prevail, which is positioned from directly above and embedded by means of compressed air jets. This system needs no dragging of the anchor to achieve maximum holding power as in conventional systems.

We shall endeavour to provide information on proprietary anchor types together with current costs as soon as possible.

We hope that this brief document is sufficient at this stage to confirm that all aspects of the transportation proposed are perfectly feasible using currently available methods. We have enclosed some sketches of the proposals and also include typical details of equipment owned by Sunter-ITM which would be used.

We much appreciated the opportunity of discussing the project with you and your colleagues, for which we are very grateful. We should like to wish you every success and look forward to being of further assistance in the near future.

Yours faithfully,

JOHN WILSON  
Technical Director
Dear Sirs,

Wave Energy Project

From your letter of 29th October and our discussions it is evident that there are three fabricated components in such a project where Whessoe’s expertise and associated facilities are relevant. However, in view of the limited and preliminary nature of the information available to us at this stage it is not possible to make accurate assessments of costs but just to give some idea of cost areas we have set indicative tonnage prices against each of the three items. These figures are based on your specified dimensions, thicknesses and weights which have not been checked and are summarised as follows:

Item 1.

Pressure vessel to contain gyroscopes and generator. This vessel would be designed to withstand an internal vacuum and an external pressure of 10 atmospheres. It is understood that the vessel will be approximately 22mm thick on the shell and will have external reinforcing straps and flat ends 55mm thick suitably stiffened and weight approximately 108 tons. Such a vessel would be fabricated and tested to one of the agreed British Standards. For the purpose of pricing we have assumed BS5500 Construction Category 2 including spot radiography, excluding stress relief.

Budget ex works price excluding any internal and external attachments other than external reinforcing straps, would be approximately £1,100 per ton at present day levels.

Item 2.

Spine Joints to couple adjoining ducks. The dimensions of these units necessitate completion of fabrication and trial assembly at our Dock Point waterside facility. It is understood that the plate thickness will be in the order of 25mm to 35mm and the weight approximately 200 tons. The degree of reinforcement and the amount of plate cut-out for lightening purposes has an influence on the price of such a fabrication as also has the accuracy specified for the ram mountings. As a budget ex works......
price (excluding provision and fitting of swivel joint, of all hydraulic gears and flexible shrouds etc. i.e. fabricated platework only) we suggest £1,500 per ton ex works at present day levels.

This provides for a certain amount of machining of small components but not for any major machining operations. We assume fabrication generally to normal commercial standards with butt welds full penetration welded to ASME VIII Standard including spot radiography, excluding stress relief.

Item 3.

Beak of a duck. Here again final assembly would have to be at a waterfront site such as Dock Point. The plate work and reinforcement we understand would be approximately 10mm thick and the total weight approximately 65 tons. The means of attachment between the body and the beak is not shown but we would expect this to incorporate substantial reinforcement which could affect the overall cost. We suggest you allow £1,000 per ton ex works at present day prices.

Fabrication standard assumed as for Item 2 above.

Notes:
1. We have assumed that there are no temperature problems which call for superior grades of steel and for pricing purposes we have based on steel to BS4360 50D throughout.

2. No allowance has been made for protective coatings or painting.

3. The pressure vessel and the beak will ultimately have to be joined together. We have not made any provision for this cost at this stage.

4. Within our industry most fabrications are individually engineered and therefore each has to carry its full engineering cost. If the engineering charge can be spread over multiple units then an obvious saving can be made up to say 5%.

However your suggested rate of production of two sets a week would call for a substantial investment on jigs and fixtures spread over many sites and we regret we cannot at this stage give any meaningful indication of the effect of this investment on the cost of individual items.

We trust that our comments are of assistance to you and we will be pleased to keep in touch as the project develops and design details become finalised.

Yours faithfully,

WHESOHE HEAVY ENGINEERING LIMITED.

F.G. Hardey
Chief Estimator - Pressure Vessels.
APPENDIX 6

RELIABILITY CONSIDERATIONS

In complex machinery with a number of functions, each dependant upon its predecessor, the failure of any one stage can cause complete failure. It is possible, however, to have a complex machine where the complexity is due to the multiplicity of items only, each being almost independent of the other. In the former case the equipment is in series while in the latter it is in parallel. An example of the former is a television set which, although reliable, can malfunction with the failure of a single component. An example of parallel operation is a telephone exchange where the failure of one line has very little effect on the overall installation.

When one considers the causes of failure of mechanical equipment they are mainly wear, dirt corrosion or metal fatigue.

In the case of the power generation equipment we are attempting to provide a design and an environment which largely eliminates these causes of failure, and is finally sealed to maintain conditions for the anticipated life. In support of the policy of sealed units one might point to an ordinary domestic refrigerator unit - a mass production item, factory sealed for a life of many years. By careful production control of the various items followed by equally well controlled assembly, a long built-in life should be attainable.

We also have in mind a control system which will monitor each of the ring cam pumps (all of which can be regarded as operating in parallel) and render inoperative any faulty units. We anticipate building in 30% over-capacity.

Even with these factors in mind it should be noted that even the failure of a complete duck only constitutes 0.1% of the total capacity.


7.1 DUCK COSTS

Basis - 1020 Ducks (510 Spines) in 10 years

Capital Costs: £ x 10^6

Electrical Transmission 320.0
Preliminaries for Civil Work 90.0
Preliminaries for Mechanical Work 30.0
Development, Tooling on sundry items, including Shore Station 25.0

Allocation per Duck: 465 x 10^6 = £456,000

1020

Direct Manufacturing Costs: £ x 10^3

Duck Bodies in Concrete (from Appendix 7.7) 68.3
Spine in Concrete: 217.4 x 1/4 (from Appendix 7.7) 103.7
Spine Joint Fabrication: 150 tons @ 600 x 1/4 45.0
Spine Joint Rams: 24 x 12,000 x 1/4 (Quote) 144.0
Spine Joint Bearing: 150,000 x 1/4 (Quote) 75.0
Spine Machinery: (2 Hydraulic Mtrs & Generators) x 1/4 31.0
Spine Machinery Canister: 10 tons @ 1200/ton x 1/4 6.0
Cabling through Spine Say 10.0
Electrical Connector £15,000 7.5
Moorings (27,000 - 2 & Anchor & Buoys & Ancillaries) 95.0
Ballasting System (Compressors, Air Bottles, Controls etc.) Say 50.0
Spine/Duck Bearings Say 20.0
Towing (Between sites and to Station) 63.3
Protective Treatment 10.0
Power Module (including Beak - see page 2) 732.0

Add Overall Contingency of 10% £1,505.8

Add Capital Allocation of £456,000 456.0

£2,112.4

*Duck price used in D.C.F. calculations.
7.2 GYRO ASSEMBLY

Total Weight: 34.46 Tons

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<tr>
<th>Component</th>
<th>Weight (t)</th>
<th>Cost (£)</th>
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<td>Ring Cam Pumps: 19.5 tons @ 3000/ton</td>
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</tr>
<tr>
<td>Rotating Assembly Disc: 17 tons @ 600/ton (x4)</td>
<td>40,800</td>
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</tr>
<tr>
<td>Other Rotating Components: 6.53 tons @ £2000/ton (x4)</td>
<td>52,240</td>
<td></td>
</tr>
<tr>
<td>Space Frame: 3.47 tons @ £600/ton (x4)</td>
<td>8,328</td>
<td></td>
</tr>
<tr>
<td>Pair of Bearings (Precessional): 1.95 tons x 3000 (x4)</td>
<td>23,400</td>
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<tr>
<td>Ring Cams 1.75 tons x £2,000 (x4)</td>
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</table>

Total: £228,388

Say £242,388

7.3 POWER MODULE COSTS

<table>
<thead>
<tr>
<th>Item</th>
<th>Cost (£)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure Vessel 108 tons @ £1200/ton</td>
<td>129.5</td>
</tr>
<tr>
<td>Leak Fabrication: 62.5 tons @ £600/ton</td>
<td>37.5</td>
</tr>
<tr>
<td>Gyro Flywheel Assemblies: (from 7.2)</td>
<td>245.0</td>
</tr>
<tr>
<td>Hydraulic Motors: 10 @ £6,000</td>
<td>60.0</td>
</tr>
<tr>
<td>Generator allowed in electrical costs by M &amp; M</td>
<td>100.0</td>
</tr>
<tr>
<td>Miscellaneous items: (Cable Reel, Piping Etc.)</td>
<td>Say 100.0</td>
</tr>
<tr>
<td>Control System Assembly</td>
<td>50.0</td>
</tr>
</tbody>
</table>

Total: £722.0

7.4 SERVICE CHARGE

Workshop support, including Labour & Supervision
200 men in 100,000 ft² building  Say £5 x 10⁶ per annum

Shore Control Station: 5000 ft² @ £20/annum & 8 men @ £20,000/pa inc. O/H and heating, electrics, telephones etc.  Say 1 x 10⁶ per annum

5 Helicopters @ £100/hr each for 100 days/year 12 hours/day  Say 1 x 10⁶ per annum

5 Tugs - 100 days/year @ £3000/day 1.5x 10⁶ per annum

10 Mooring Layers and attendance  Say 100 days @ £5000/day 5 x 10⁶ per annum

Sundries 5 x 10⁶ per annum 18.5x 10⁶

Daily Service Charge is therefore:

\[
\frac{18.5 \times 10^6}{365.4} = \frac{50,643}{\text{per day}} \quad \text{or } \frac{1}{4} \text{ of Capital Cost} \quad (1.09\%)
\]
## 7.5 GYRO ASSEMBLY

### WEIGHT

<table>
<thead>
<tr>
<th>Description</th>
<th>Kg</th>
<th>Tons</th>
</tr>
</thead>
<tbody>
<tr>
<td>ROTATING ASSEMBLY (inc. disc) from E.U. Specification</td>
<td>23873</td>
<td>23.53</td>
</tr>
<tr>
<td>SPACE FRAME</td>
<td></td>
<td></td>
</tr>
<tr>
<td>12 x 0.282 (36 x 10 x 4 x 0.5)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(30 x 1 x 6 x 3)</td>
<td>6264</td>
<td>1931</td>
</tr>
<tr>
<td>Plates</td>
<td></td>
<td></td>
</tr>
<tr>
<td>12 x 0.282 (15 x 15 x 1 x 2)</td>
<td>1523</td>
<td>689.7</td>
</tr>
<tr>
<td>Nodes</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2 x 12 x 12 x 10 x 0.282</td>
<td>812</td>
<td>184</td>
</tr>
<tr>
<td>BEARING (STATIC) - GYRO</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2 x 0.282 ($\frac{11}{2}$ x 51 x 27.6)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>- ($\frac{11}{2}$ x 39.37 x 19.7)</td>
<td>18274</td>
<td>8306</td>
</tr>
<tr>
<td>4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>BEARINGS (PRECESSIONAL)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2 x (180 Kg + 390 Kg) (Makers Weights)</td>
<td>1104</td>
<td>0.5</td>
</tr>
<tr>
<td>HOUSING</td>
<td></td>
<td></td>
</tr>
<tr>
<td>.282 ($\frac{17.5}{2}$ II x 21.6)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>- ($\frac{17.25}{2}$ II) x 9.8 x 2</td>
<td>7024.5</td>
<td>3181</td>
</tr>
<tr>
<td>4</td>
<td></td>
<td></td>
</tr>
<tr>
<td>RING CAMS (STATIC)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>157.5 x II x 7.86 x 3.94) x 2 x 0.282</td>
<td>8576.3</td>
<td>3884</td>
</tr>
<tr>
<td>= 8576.3 lbs</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

|                         |      |      |
|                         | 32.27|      |
Initial investment for setting up project and development cost = £y

Each duck costs £x in year 0

Total number of ducks = 3,330

No. of ducks built in year

<table>
<thead>
<tr>
<th>Year</th>
<th>Ducks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>a</td>
</tr>
<tr>
<td>2</td>
<td>b</td>
</tr>
<tr>
<td>3</td>
<td>c</td>
</tr>
<tr>
<td>4</td>
<td>d</td>
</tr>
<tr>
<td>5</td>
<td>e</td>
</tr>
<tr>
<td>6</td>
<td>f</td>
</tr>
<tr>
<td>7</td>
<td>g</td>
</tr>
<tr>
<td>8</td>
<td>h</td>
</tr>
<tr>
<td>9</td>
<td>j</td>
</tr>
<tr>
<td>10</td>
<td>k</td>
</tr>
</tbody>
</table>

Operating and maintenance costs = £z per year

Cost per MW year of electricity in year 0 = £w

Inflation = r%

Interest = i%

Each duck produces approximately 0.6 Mw on average.

The total average output = 2,000 Mw

Total project life = 25 years

Initial investment is incurred in Year 0

Ducks production in year 1

Ducks production in year 10
CASH FLOW DISCOUNTED BACK TO YEAR 0

<table>
<thead>
<tr>
<th>YEAR</th>
<th>CASH FLOW</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>(-y)</td>
</tr>
<tr>
<td>1</td>
<td>(-ax \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>2</td>
<td>(-bx + 0.6aw - z) \left/ \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>3</td>
<td>(-cx + (a + b)0.6w - z) \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>4</td>
<td>(-dx + ((a + b + c)0.6w) - z) \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>5</td>
<td>(-ex + ((a + b + c + d)0.6w) - z) \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>6</td>
<td>(-fx + ((a + b + c + d + e)0.6w) - z) \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>7</td>
<td>(-gx + ((a + b + c + d + e + f)0.6w) - z) \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>8</td>
<td>(-hx + ((a + b + c + d + e + f + g)0.6w) - z) \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>9</td>
<td>(-jx + (((a + b + c + d + e + f + g + h)0.6w) - z) \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>10</td>
<td>(-hx + (((a + b + c + d + e + f + g + h)0.6w) - z) \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>11</td>
<td>((2000w - z)) \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>12</td>
<td>((2000w - z)) \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>24</td>
<td>((2000w - z)) \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
<tr>
<td>25</td>
<td>((2000w - z)) \left( 1 + \frac{r}{100} \right) \left/ \left( 1 + \frac{s}{100} \right) \right.\</td>
</tr>
</tbody>
</table>
Assume:

\[ y = 14.1 \times 10^6 \]
\[ x = 1.6 \times 10^6 \]
\[ a = b = c = 102 \]
\[ r = 0 \]
\[ S = 10\% \]

\[ Z = 50,000 \text{ day} = 15.25 \times 10^6 \text{ ann.} \]

Pay back time 3.5 years
Year 0  
\[- \£ 410\]

Year 1  
\[
\left[ -102 \times 1.16 \times 10^6 \right] \times 0.9092 \\
= \ - \£ 148.3
\]

Year 2  
\[
\left[ -102 \times 1.16 \times 10^6 - 18.25 + 0.42 \times 1.02 \right] \\
\times 0.9092 \\
= \ - \£ 149.9 + 35.4W
\]

Year 3  
\[
\left[ 181.45 + 0.42 \times 1.02 \times 204 \right] \times 0.9092 \\
= \ - \£ 136.2 + 64.4W
\]

Year 4  
\[
\left[ 181.45 + 0.42 \times 1.02 \times 306 \right] \times 0.9094 \\
= \ - \£ 123.9 + 87.78W
\]

Year 5  
\[
\left[ 181.45 + 408 \times 0.42 \times 0.9095 \right] \\
= \ - \£ 112.6 + 106.4W
\]

Year 6  
\[
\left[ -181.45 \times 510 \times 0.42 \right] \times 0.9096 \\
= \ - \£ 102.4 + 120.9W
\]

Year 7  
\[
\left[ -181.45 \times 612 \times 0.42 \times 0.9097 \right] \\
= \ - \£ 93 + 131.9W
\]
Year 8
\[-181.45 \times 714 \times 0.42 W \times 0.909^8 \]
\[= -£ 84.6 + 139.89 \]

Year 9
\[-181.45 \times 816 \times 0.42 W \times 0.909^9 \]
\[= -£ 76.9 + 145.35 \]

Year 10
\[-181.45 \times 918 \times 0.42 W \times 0.909^{10} \]
\[= -£ 69.9 + 148.65 \]

Year 11
\[-18.25 + 1026 \times 0.42 W \times 0.909^{11} \]
\[= -£ 6.4 + 150.15 W \]

12-25 = -£ 47 + 1105 W

Total = -1561.1 + 2235 W

Equating to Zero
\[W = £ 0.70 \times 106 \]

7.97 \( \)
**Assumptions:**

- \( y = £ 465 \times 10^6 \)
- \( x = £ 1.6 \times 10^6 \)
- \( a = b = c = 102 \)
- \( r = 0 \)
- \( S = 5\% \)
- \( z = 50,000 \) day = \( £ 18.25 \times 10^6 \) per annum

Payback time 25 years
Year 0
\[-4.65 \times 10^6\]

Year 1
\[-102 \times 1.6 \times 10^6 \times 0.952^2\]
\[= -1.155 \times 10^6\]

Year 2
\[-102 \times 1.6 \times 10^6 - 18.25 + 0.42W \times 102\]
\[\times 0.952^2\]
\[= -181.45 \times 10^6 + 42.84W\]
\[\times 0.952^2\]
\[= -1.64 \times 10^6 + 38.83 W\]

Year 3
\[-181.45 \times 10^6 + 0.42W \times 204\]
\[\times 0.952^2\]
\[= -1.156 \times 10^6 + 73.92 W\]

Year 4
\[-181.45 \times 10^6 + 0.42W \times 306\]
\[\times 0.952^4\]
\[= -1.49 \times 10^6 + 105.56 W\]

Year 5
\[-181.45 \times 10^6 + 0.42W \times 408\]
\[\times 0.952^5\]
\[= -1.141.89 \times 10^6 + 134 W\]
Year 6

\[-181.45 \times 10^6 + 0.42W \times 510 \times 0.952^6\]

\[= \frac{-1135.08 \times 10^6 + 159.46 W}{128.59 \times 10^6 + 182.16 W}\]

Year 7

\[-181.45 \times 10^6 + 0.42W \times 612 \times 0.952^7\]

\[= \frac{-128.59 \times 10^6 + 182.16 W}{125.2 \times 10^6 + 202.32 W}\]

Year 8

\[-181.45 \times 10^6 + 0.42W \times 714 \times 0.952^8\]

\[= \frac{-116.54 \times 10^6 + 220.13 W}{110.95 \times 10^6 + 235.76 W}\]

Year 9

\[-181.45 \times 10^6 + 0.42W \times 816 \times 0.952^9\]

\[= \frac{110.95 \times 10^6 + 235.76 W}{10.62 \times 10^6 + 249.38 W}\]

Year 10

\[-181.45 \times 10^6 + 0.42W \times 918 \times 0.952^{10}\]

\[= \frac{110.95 \times 10^6 + 235.76 W}{10.62 \times 10^6 + 249.38 W}\]

Year 11

\[-18.25 \times 10^6 + 0.42W \times 1020 \times 0.952^{11}\]

\[= \frac{10.62 \times 10^6 + 249.38 W}{10.62 \times 10^6 + 249.38 W}\]
Years 12-25

\[-18.25 + 428.4W \left( 0.952^{12} + 0.952^{13} \right) + \ldots + 0.952^{25} \]

Sum of terms in brackets ( ) can be expressed as the sum of a geometric progression.

\[ S = 0.952^{12} \left( 0.952^{14} - 1 \right) \]
\[ 0.952 - 1 \]
\[ = 0.952^{12} \left( 10.37 \right) \]
\[ = 5.75 \]

\[ \therefore \text{Years 12-25} \]
\[ = - £ 104.94 \times 10^6 + 2463.3 \text{ W} \]

Total
\[ - £ 1961.47 \times 10^6 + 4064.82 \text{ W} \]

Equate to Zero
\[ : W = 0.4825 \times 10^6 \]
\[ = 5.5 \text{ } \]
TO: Mr. P.B. Williams
DATE: 30th November 1979
OUR REF: DCK/DJ

SUBJECT: 100100 - WAVE ENERGY INSTALLATION

PL00100/1A - Overall Programme - Design, Procure, Construct, Installation

I attach six copies of the above programme for inclusion in your report.

Please advise me if you require any changes to be made to the programme and/or if you require additional copies.

D.C. KILPATRICK

CC: Messrs: J.F. Mayer
J.P.F. Cooke
D.M. Preedy
File
EDINBURGH-SCOPA-LAING

LAING

5TH YEAR

WAVE ENERGY

REPORT

APPENDIX S

"RECENT PROGRESS ON DUCKS" PAPER BY S.H. SALTER
SUMMARY

Until the end of 1978, the team at Edinburgh put most of its effort into small scale laboratory tests with increasing levels of hydrodynamic realism. Some test results show that the requirement for rigidity of the duck back-bone is much lower than was predicted and that in waves with lengths of twenty diameters and more there are considerable benefits to be derived from controlled back-bone movements.

Since the beginning of 1979 we have been working on the problems of full-scale design. The officially preferred schemes of the British programme use low pressure air turbines, asynchronous generation, DC transmission, serial connections and simple designs with reliability achieved by easy access and maintenance. We are out of step and prefer high pressure oil hydraulics, synchronous generation, AC transmission and parallel connections. We exploit the techniques of modern electronics and plan to achieve reliability by total hermetic sealing of the power conversion mechanism which will exclude all organisms, both marine and human. I shall attempt to justify our approach.
MOVING AXIS RESULTS

Several authors, Count, Glendenning, Mei, and Standing have published theoretical efficiency curves for ducks and suggest that movement of the mounting reduces the performance.

It is true that the early ducks were intended to operate on a relatively stable reference frame and movement was regarded as necessary for survival. But it is not true to suppose that there is a continuous degradation of performance from absolute rigidity to total freedom. The experimental evidence is more complicated, more interesting and more promising than the straight degradation theory would suggest.

Our mounting allows separate control of the rigidity of the model in heave and surge. We made efficiency measurements at a number of frequencies and a number of combinations of rigidity. A typical result is shown in Figure 1.

The horizontal axis is compliance in the surge direction and the vertical is compliance in the heave direction. The wavelength corresponds to 22 duck diameters and the curves show contours of efficiency. There are two separate regions of good performance and neither occurs at the point of absolute rigidity. They are separated by a region of extraordinarily low efficiency. We call this area 'death valley'. It is a feature of tests at all frequencies, and may provide a useful operating condition for survival.

At high frequencies, with wavelength say eight times the diameter of the duck, then the pattern of the contours moves closer to the point of total rigidity and at longer waves the contours move towards the more compliant area. This leads to significant improvements in efficiency in very long wave conditions. Figure 2 shows the effect of varying heave compliance with the surge axis at a constant compliance of $4.5 \times 10^{-6} \text{ m}^2/\text{N}$. The wave period is twelve seconds.

It is fortunate that ducks prefer rigid mountings for short waves and floppy ones for long waves because we ought to expect that the crest lengths found at sea will be related to wave lengths. It is fortunate
too that the rigidity of concrete tubes is more than is required for most of the expected conditions.

It may be possible to explain the phenomena as follows. In short waves the water movement is concentrated near the surface and very little energy passes beneath the duck so that it operates in the way expected. But at longer waves there is movement well below the duck's draught and it becomes more difficult to attract it to the duck instead of to the water to leeward. But if the stern of the duck can be moved in such a way as to generate waves which are the inverse of those sneaking below then the water behind must be calm. The back of the duck is behaving like an Evans' cylinder and, indeed, it was the work of David Evans which led to the discovery of this effect.

If the techniques of reactive power take-off and compliant mounting are exploited, it is possible to achieve, with a 10 metre diameter duck, the efficiency curve of Figure 3.

We find that the frequency axis used for the presentation of mathematical results is too cramped at the low frequency end and masks commercially important improvements. In Figure 3, we have used a period axis but varied the intervals between periods in proportion to the power content of the South Uist wave climate below one metre RMS. This corresponds to a reasonable guess of the sensible power limit.

THE BEARING BETWEEN DUCK AND BACKBONE

Forces

Both the fixed and the moving rig are fitted with strain gauge sensors from which are calculated the surge and heave forces acting on the models. A great deal of the tank work has been concerned with the measurements of these forces in both ducks and cylinders.

The established theory for fluid loading was developed by Morison and intended for use on vertical submerged cylinders. Our models are horizontal and pierce the surface. It has not been easy to relate the
results to theory. Some unusual effects are caused by the interplay of
buoyancy and the vertical inertial force which results in downward forces
during both the crest and the trough of the wave. A reconciliation has
been achieved by Dixon. (6) We have found that the forces on ducks can
be reasonably predicted on the assumption that they are proportional to
wave height and duck diameter, the density of water, the acceleration of
gravity and a single all-embracing force coefficient measured for the
wave conditions concerned. The variation of force coefficient with
wave length and steepness is not large. A typical value for surge is
0.4 and for heave 0.25 with a tendency to fall in larger waves. The
use of a force coefficient brushes aside all the niceties of fluid
loading theory but it gets reasonably accurate answers fast.

If we attempted to hold a duck rigid, the extreme waves in the open
Atlantic would raise forces between 150 and 200 tons for every metre
of duck width. But the attempt to hold anything rigid against these
waves is not an intelligent or productive activity. We conducted
a series of experiments on the moving rig in which we maintain rigidity
up to a limit of force and then allow the rig to be deflected at a
constant force. It seems that the economically justified limit for
inshore equipment is about 10 tons per metre. But if the duck is
allowed to yield to forces higher than 10 tons per metre then it is
still necessary to provide a force to accelerate the backbone. Until
we have data from a model string with the correct values of compliance
we are unsure of the value of this acceleration. The narrow tank
results suggest that it will lie between 0.5 and 1.0 g. As the back-
bone will weigh 75 tons per metre, the forces needed to accelerate it
are up to one half of those needed to hold it stationary. For a long
time we thought that this acceleration requirement raised difficulties
in the design of the bearing surface between duck and backbone.
Fortunately the reasoning was fallacious.

Archimedes to the Rescue

The duck's back-bone is neutrally buoyant and floats inside it. Its
weight is balanced by its buoyancy. The buoyancy force is the back-bone
volume times the density of water times the acceleration of gravity.
Transporting the apparatus to another planet might alter the acceleration of gravity but would not induce a force between duck and back-bone, which would still float as before with back-bone weight and buoyancy force in balance. The action of waves produces temporary alterations in the direction and value of gravitational acceleration and the water between back-bone and duck produces temporary alterations in the direction and value of the buoyancy forces to maintain the balance as before. The water wants to be left behind just as much as the back-bone. It would be foolish as well as impossible to try to keep the space between duck and back-bone dry.

The effect may be demonstrated by shaking a jar of pickled onions, but it is also interesting to consider an experiment on an object with excess buoyancy such as a hydrogen balloon in a car. If the car is accelerated, the balloon moves to the front because the heavier air claims the right to be left behind. When the car is braked, the balloon moves to the back.

**Bearing Requirements**

While it may be prudent to provide for occasional overload forces of say 100 tons per metre, caused by handling and collision, there is no need for the normal operating forces to exceed about 10 tons per metre. Despite the fact that loads are not shared evenly over the projected area of a bearing, the value of force over projected area is a convenient parameter for comparison. With 10 tons over 9.8 square metres, we have about $10^4 \text{ N/m}^2$ or less than 1.5 psi. This is an extraordinarily low value. It is difficult to find examples in bearing technology which are as low. The maximum angular velocity between duck and backbone will be less than 5 metres per second (11 mph) and for most of the time only one tenth of that.

So far the specification is by no means very demanding but it turns out that the fraction of power lost by the bearing is about double the value of the coefficient of friction. This means that we have to achieve much lower friction than is usually expected of plain bearings. Furthermore, friction and wear are associated. All the economic
considerations force the design towards a much longer working life than would be expected for land based plant.

Some very successful water lubricated rubber bearings are made by BTR. They are known as Silverline and are shown in Figure 4. The shape of the rubber entraps a water film and so the coefficients of friction and rates of wear are exceptionally low. They are widely used for the stern-tube bearings for ships. Unfortunately, the hydrodynamic effects work only at peripheral speeds which are too high for our application. When the duck motion is reversed, the film would be lost and the friction would rise.

The section moment of a neutrally buoyant tube depends on the fourth power of its diameter. This means that the structural demands of the back-bone need every millimeter of diameter. We cannot spare the radial distance needed for a rolling tyre bearing although the dark, wet, slow, cool conditions and absence of braking and cornering forces provide ideal conditions for rubber tyres. Another difficulty is that we cannot expect back-bones to be round or parallel. Errors of \( \pm 15 \text{ mm} \) would be likely even with the very best civil engineering.

To summarise: we need to design a radially thin bearing for low loads, moderate but reversing velocities, very low friction and zero wear, which can work in sea water and can tolerate large geometrical errors.

**Slubber**

Slubber is the general name for an elastomeric body which, when subjected to external pressure, exudes a liquid. One way in which this can be achieved is shown in Figure 5.

One side of the bearing is made from concrete clad with a thin skin of cupro-nickel. This provides a smooth, if inaccurate, surface. Riding on top of the cupro-nickel is a circular pad like a hovercraft. But instead of having a pressure supply from an air turbine, the pad is fed with water from a second chamber with corrugated bellows for walls. If this second chamber has a diameter less than the hovercraft pad, the
pressure inside will be greater and a restrictor between the bellows chamber and the pad will, in combination with the impedance of the gap between the pad and the cupro-nickel, govern the rate of flow of water out of the bellows chamber. The hovercraft will fly over the surface of the cupro-nickel with negligible friction and wear.

An alternative scheme uses a bellows chamber marginally greater in diameter than the pad. This means that a small fraction of the load is taken by pad lands but the leakage rate is much reduced. This arrangement could be regarded as a friction coefficient attenuator rather than a true hydrostatic bearing.

Either type will work only for loads which alternate. The bellows chamber must be recharged during the portion of the cycle when the load is removed. But this alternating loading is exactly what happens with waves.

The choice between the zero contact and the friction attenuating type will depend on how well we can make the pad lands conform to the contour of the skin.

Slubber bands will run round the annular space between duck and backbone. Slubber could be mistaken for the skin from the tentacles of a giant octopus suffering from a mutation which causes it to blow rather than suck.

**FULL SCALE BACK-BONES**

The function of a duck's back-bone is to provide a reference by connecting it to waves of opposite phase. This is a more difficult task than to couple it to inertia down below in the calm water like the oscillating water column or to go backwards like the rafts. But the mechanism used to connect to the reference is itself a useful energy producing plant and we judge the attempt to be worth making. It allows us to concentrate arbitrarily large amounts of power and avoids side to side collision difficulties.
We realised very early that bending moments and shear forces would dominate the design of the back-bone. Whatever rules for fluid loading are used it is clear that brute strength offers no solution.

Experiments on very long compliant pipes suggested that the bending moments were lower in the central regions of the back-bone and so we are designing for 'infinitely' long strings. It is necessary to let the back-bone deflect so as to yield to waves with amplitudes greater than the ducks wish to absorb. I believe that proper control of the joints will prove an extremely valuable technique.

Our present design uses pairs of ducks on 60 metre lengths of concrete back-bone joined to the neighbours with Hooke's joints. The motion of the joints is controlled by double acting hydraulic rams arranged around the circumference of the back-bone section. The maximum angle is ± 12°. Sets of hydraulic rams on opposite sides of the backbone are cross-connected and drive a variable displacement axial piston swash plate motor/pump of the same type as is used in the power take-off.

The hydraulic motors drive synchronous motor/generators running in phase with the rest of the electrical network. The joints can act as a power generating mechanism in their own right. This is invaluable for starting gyros. They can be used to determine the compliance of the back-bone with different values for different frequencies. They can let the back-bone yield to extreme waves but restore the mid-position afterwards. They can damp out flexure oscillations. It is not stretching imagination too far to propose a sequential sinuous series of torque commands which give sections of duck string their own eel-like mobility without tugs.

If we use the naval architect's rules of thumb for drag estimates, it seems that speeds of one or two knots require remarkably small amounts of drive power. Duck strings can be self-propelled from wave energy while duck-masters sit comfortably at home watching data relayed from satellites. All these benefits can be realised by the control of the angle of a swash plate motor. I dare say that quite an interesting box of electronics will be needed to work out the angles but modern electronics is very good at making interesting boxes.
A large area of uncertainty surrounds the choice of joint angle. We know much less about the crest length of waves than about their amplitudes and periods. We know very little about the behaviour of non-linear beams subjected to wave loading. Testing our new model will keep us fully occupied.

I believe that we will not find problems with the design of the full-scale joint bearings. Glacier DX linings promise excellent pressure velocity properties and the reversing direction of the loading is a valuable asset allowing continued regreasing. However, it looks as though presently available ram technology will not offer lifetimes comparable with the concrete and power take-off mechanism. Long life rams may need further development.

POWER CONVERSION

Working Fluid

As soon as the wave power investigator begins to test in the realistic spectra of irregular waves, he is dismayed to discover the very wide range of instantaneous power densities. If we are to make electricity, this spiky flow of dispersed energy must be concentrated, rectified, regulated and transformed to a high velocity low torque unidirectional motion. This series of processes is best performed in a fluid and the majority of proposals for wave energy devices use a fluid interstage.

We can choose to use air, water or oil. Many people in the British wave energy programme are keen on air systems. They argue that despite the low pressures involved it is possible to achieve high velocities and even rectification in a single leap. They argue further that the air turbines are 'simple' with only a single moving part. I have been under some compulsion to adapt ducks to use a low pressure air system. I have resisted for the following reasons:

1. Mechanisms which use air in free exchange with the atmosphere are not using a clean dry mixture of oxygen and nitrogen. There will be water vapour, rain and spray. There will be crystals of salt
and ice. There will be all the flotsam of the sea including weed, beer cans, logs and carelessly low-flying seagulls. We have to avoid obstructions in the air passages but be very sure that solid chunks of water are kept away from the rapidly moving turbine blades. The open air and natural sea water are ill defined working media.

2. It is possible to design devices with their own private system of recirculating air from which these hazards are excluded. The French air bag is an excellent example. The air is contained by a flexible skin but this puts an immense responsibility onto the designer of the fabric. Any failure can result in a sudden loss of buoyancy. Reliable damage control systems, which may need mechanical moving parts, must be provided.

3. The efficient operating band of the air turbines so far invented is narrow and the efficiency falls at the low power end when we need it most. It would be possible to improve things by means of variable pitch blades and moving guide vanes but these would at once lose the advantages of simplicity claimed for the single moving part.

4. It is, in any case, not true that an air turbine has only a single moving part; an apparent simplicity masks a dangerous complexity. Each blade of the turbine can have many modes of vibration, each coupled to one another, and the surrounding structure. Low drag demands thin foil sections while rigidity demands fat ones. The blades are moving at high speed in a fluttery medium with very little damping. Changes of speed and air velocity will search out every possible mode of vibration. Curing one will induce another.

I like my complexities to be apparent. I like my working conditions exactly defined. I want the chance to exploit clever control. I want a clean working fluid with the most benign properties that I can achieve. That fluid is high pressure oil.
Problems for Ducks

The problem of converting the motion of a duck into electricity has proved an interesting challenge. The most obvious course would be to build a pumping system driven by the relative motion between duck and back-bone. The lack of geometrical precision would make the use of gear driven pumps difficult but tapes and toothed belts could stand the necessary tension and allow for generous tolerances. But because of the Archimedes bearing force argument, it would be foolish to attempt to keep the space between the duck and back-bone dry and so any pump design would require a seal to keep out sea water. The life of this seal has proved the stumbling block. We could only expect a few years of operation, and this is not acceptable.

The Power Take-Off That Came In From The Wet

It has turned out to be possible to design a power take-off system which works in completely sealed conditions using a scheme which provides many other advantages. Indeed, our only problem has been that physical arguments and mathematics are not sufficient to persuade critics. The feel of a working model is necessary. It is as if a magic emanation radiated from it. To touch is to believe.

The scheme exploits the behaviour of the gyro. I am indebted to Professor E R Laithwaite for the suggestion. Thinking about gyros is difficult for two reasons. The first is that we need to imagine torques and angular velocities which are harder to visualise than their linear counterparts. The second is that we have to think in three dimensions.

The axis of spin of a gyro defines one direction. We can consider the two other axes perpendicular to the spin axis as the 'ports' of a transforming device which converts torque into angular velocity. The gyro does not know which port we are labelling as an input and the exchange of torque for angular velocity works both ways. Input torques produce output angular velocities and output torques require input angular velocities. If the port we choose to label an output is moving without opposition, then there will be torque but no angular velocity at
the input. Now a device which has a torque in one place and an angular velocity at another could also be described as a gear box with an infinite ratio of speed increase. Speed increasing gear boxes, especially efficient ones with high ratios, are very difficult things to make and are just what wave energy needs.

Our gyros are fitted in a canister in the beak of the duck. Its diameter can be about half the back-bone diameter. We use pairs spinning in opposite directions about axes initially perpendicular to the duck's axis of nod. When the waves apply a torque about the duck's axis of nod, the gyros will precess about the third axis perpendicular to both the axis of spin and nod. This precession movement can be used to do work if it is opposed by a torque. It is necessary for there to be a torque on the output if the duck is to move, and this output torque will determine the angular velocity of nod. Because a pair of gyros spin in opposite directions they will also precess in opposite directions and so the two output torques will be in opposite directions and can be cancelled by stresses in the duck without bothering the back-bone.

The 'gyroness' of a gyro is determined by the two parameters $I_\omega$ which remain together in all the equations. $I$ is the moment of inertia of the disc and $\omega$ is its angular velocity. If $\phi$ is the angle to which the gyro has precessed, the torques and velocities are related as follows:

\[ T_{\text{out}} = I_\omega \Omega_{\text{in}} \cos \phi \]

\[ T_{\text{in}} = I_\omega \Omega_{\text{out}} \cos \phi \]

\[ \frac{T_{\text{in}}}{\Omega_{\text{in}}} = I^2 \omega^2 \cos^2 \phi \frac{\Omega_{\text{out}}}{T_{\text{out}}} \]

But $\frac{T_{\text{in}}}{\Omega_{\text{in}}}$ is the hydrodynamic damping coefficient of the duck. Let us call it $\kappa$. We must get it about right if the duck is to behave correctly in the water.

Everything is inverted when seen through a gyro. Low damping coefficients on the output look like high damping coefficients on the input. Captivity looks like freedom. Similarly, spring and inertia are interchanged, with big inertias looking like low rate springs. Death valley looks like the garden of peace.
The \( \cos \phi \) term reduces the usefulness of the gyro by less than one at first expects. We require the greatest amount of torque and power at the central position of the system. It would be sensible to arrange that the biggest waves in a sea state drove the gyros through nearly the maximum angle of 180°. We can arrange this by altering the speed of gyro spin. This will set a number \( r \) which is rather like a gear ratio.

\[
\begin{align*}
  r &= \frac{\omega_{\text{out}}}{\omega_{\text{in}}} = \frac{T_{\text{in}}}{T_{\text{out}}} \\
  \text{and } I^2 \omega^2 \cos^2 \phi &= \frac{T_{\text{in}}}{\omega_{\text{in}}} \cdot \frac{T_{\text{out}}}{\omega_{\text{out}}} \\
  &= \frac{K}{r^2} \\
  \therefore \quad I\omega &= \frac{K}{r^2 \cos \phi}
\end{align*}
\]

As the angle of precession increases, the value of \( r \) rises.

If we reduce the speed of gyro rotation we increase the value of the gear ratio. In calm conditions we expect to run at a midpoint gear ratio of about six and near the power limit at a ratio of about three. But we are still at an early stage in the optimising process and many factors influence the decision. I believe that wave energy devices with larger damping coefficients could make use of even higher gear ratios. (We have carried out tank tests with ratios of up to twenty.)

**From Precession Into Oil**

The energy in the precession movement of the gyro is turned into oil pressure by a modified version of the ring cam pump. These units are made by MacTaggart-Scott, Poclain and Häggelund. Figure 6 shows the shape of a conventional ring cam with inward facing lobes driving rollers in a radial direction. The cost of such a unit depends on the sum of the cost of the lobes and the followers and so ought to rise with diameter. However, the volume pumped depends on the product of the number of lobes and cams and so rises with the square of diameter. We should therefore go toward the largest diameter that can fit inside the
power canister. But as the diameter rises the force from each roller has to travel further round the rim before it can be balanced by another force on the opposite side. For very large diameters, this becomes less attractive and so we are using pairs of rollers working against each other through the thickness of the ring, as in Figure 7. To make best use of the space inside, we twist the ring so that the rollers no longer move in its plane. It is now more like a belt than a shape cut from plate. See Figure 8.

If we think of the gyro assembly as occupying a sphere like the world with its precession bearings at the north and south poles, then the largest possible ring cam would be the equator. But the equator is a busy place. We have to crowd it with the high speed gyro bearings and their motors. So in Figure 9, we have moved the ring cams to the tropics and have the chance to fit two to each gyro. This means that they have to have a tapered section and their followers have to be slightly conical. At first we were daunted by the prospect of machinery cam lobes on a conical surface, but it turns out that the specialised machine tool required will be cheaper than that needed for a flat plate cam of comparable diameter. It can be built with rotating bearings only and needs no slideways.

The forces on the ring cams are largely compressive so that we can make them from flame-hardened cast iron. The line of rolling between cam and follower is the only place in the entire system where moving metal touches metal with a force across the contact. It is the only place in the power chain with a finite life. Fortunately, the mechanics and metallurgy of rolling contacts have been the subject of intensive study by the rolling bearing industry. We know how to calculate the fraction of a population which will survive particular loads and can make this fraction acceptably high over the design life of 25 years. Some rollers or cam lobes will fail and we shall need to provide for their early retirement.

Each roller must be coupled to a piston. This is done by a link shown in Figure 10 which gives yet another speed increase and provides restraint against unwanted roller movements. It is very important that the force should be evenly distributed along the line of contact between the roller
and the cam. Imperfections in the geometry are accommodated by a spherical bearing at the pivot point of the link and a ball-ended connecting rod between link and piston. The rollers are kept in line on the cam by small flanges. These dominate, and everything else must be free to comply. The ratio of roller circumference to cam lobe wavelength is chosen to spread the wear patches evenly round the roller circumference.

**Cam Profile**

The sinusoid is not the best form for the profile of our cam lobes. It has sharp curvature at the crest which leads to higher compressive stresses. It has high accelerations at the crest which must on no account overcome the boost pressure and let the following rollers lift. While the profiles chosen for most conventional motors are designed to give constancy of torque with acceptable life, we have so many lobes and followers that smoothness of output is not a problem. Instead we shall design for maximum life with a sharp concave curvature at the trough (nearly the same as the roller diameter) and a blunt curvature at the crest.

It may prove desirable to use a form with some asymmetry about the vertical to the cam surface. This is because the rollers on trailing links have a slightly easier time than those on leading links, and we are trying to spread the duty as evenly as possible.

The rollers are thick walled cylinders and run on hydrostatic bearings which support them evenly behind their contact area. Lubrication for the roller and pivot bearing is supplied from the working cylinder via a hole through the middle of the connecting rod. This means that the pressures are always appropriate whether the cylinder is operating or idling.

It is important that the torque which resists precession should be reflected through the gyro to give the duck the correct behaviour. This precession torque is set by the number of pistons allowed to pump. Each cylinder has an electronically controlled poppet valve which is
held open by the field of a permanent magnet. Boost pressure in the lower pressure manifold drives oil into the cylinder during the down stroke of the piston. If the inlet valve remains open then the up-stroke of the piston will return this oil to the low pressure manifold and no work will be done. But if an electrical pulse is applied to the coil controlling the inlet valve then the cylinder is enabled, the inlet valve is allowed to close and the oil raised to a high pressure \(20 \times 10^6\) N/m\(^2\). It passes through the outlet poppet into the high pressure manifold. If this pressure is constant (and I shall show later why it is constant) the torque will depend on the number of valves enabled. The total number is large enough for the gradations of torque to be essentially smooth.

The decision about which valves are to work is made by a computer to which is fed information about the duck's angle, velocity, pressure field, recent history of power levels, gyro disc speed, gymbal frame angle, condition of the mooring system, deflection of the back-bone, demand on the grid, Rotterdam spot prices, yellowcake futures, news of unrest in Namibia, and whatever other pieces of hydrodynamic or economic data may turn out to be relevant in the future.

An interesting example of the way in which microprocessor technology can revolutionise mechanical design is the use of the computing system to nurse weak points on the ring cam pump. Rolling contacts have a finite probability of failure. The earliest sign will be a change in the noise made by the rollers. A number of piezo-electric transducers around the ring cam will sense the time of arrival of fault noises and so pinpoint their source. If the blemish is on a roller then that roller can be avoided in the future. If it is on the ring cam then rollers will be off-loaded as they are due to pass over it. The maximum torque of the ring cam pump will decline with age, but it can start with a generous margin of spare capacity.

**Keeping the Pressure Constant**

A simplified hydraulic circuit is shown in Figure 11. The oil from the ring cams will drive a swash plate motor at each end of a gyro shaft in
parallel with another driving an electrical generator. If the pressure tends to rise as a result of a burst of wave energy the angular deflection of the swash plates will increase so as to allow the extra energy to speed up the gyro flywheel. If the flow of energy from the ring cam is at exactly its mean value the gyro swash plate motors will move to their zero displacement angle and all the oil will flow to the motors driving the electrical generator. If, during a lull, there is less oil from the ring cam pump than is necessary for the generator drive then the swash plates on the gyro motors will swing over so that they pump and draw the energy deficit from the gyro disc. The gyro-drive swash plates will move at about twice wave frequency.

The moment of inertia of the flywheel is such that only very small speed variations are necessary to stabilise the generator output. At the nominal speed, each duck stores about half a megawatt hour. This may be doubled if necessary.

The swash plate of the motor driving the generator is controlled so as to keep the phase of the generator correct. The phase angle determines whether it motors or generates. The inertia of the swash plate is so small and the forces available to control its angle are so large that its frequency response is very high. The generators can respond to power variations in a few milliseconds. The reserves of energy instantly available from the flywheels make for better control characteristics than any land-based system, whether steam or hydro. The entire duck string constitutes a spinning reserve capable of stabilising the grid rather than causing it problems. Whenever a pumped storage scheme is proposed, the generating boards emphasise the value of spinning reserve which is said to be worth hundreds of millions of pounds a year even if never used. The ducks are claiming those millions for being a short-term but instantaneously responsive storage scheme. They will leave pumped storage to do its proper job of overnight working.

The vast amount of flywheel storage means that every piece of electrical equipment from the shaft of the generator to the land connector can now be rated at its mean rather than its peak rating. We can deliver some of the electricity when the consumer wants it rather than when the waves provide it. There is an enormous difference between the value of a peak
and a base kilowatt hour.

**Swash-plate Motors**

The scheme described above demands exemplary performance from the swash-plate motors. The design selected for ducks was developed by Clerk for flywheel energy storage applications. It is well described in reference (7) but its design is so remarkable that some of its features should be mentioned in this paper.

Most hydraulic pumps and motors run with their casings full of oil. While this serves to lubricate any areas overlooked by the designer, churning losses limit the speed to about 300 rpm. The Clerk motor runs with its casing evacuated. This has several advantages. Churning is avoided. Vacuum stripped oil gives fewer cavitation problems. There is no degradation of the oil from oxidation. There is no problem about sealing the drive against the vacuum of the duck flywheel enclosure.

Several designs of swash-plate motors allow side forces from the plate to spoil the contact between cylinder and piston. The Clerk design provides ball-ended connecting rods so that no side loads develop.

Most designs have cylinders which increase in diameter under applied pressure. Clerk has sealed his cylinder liners to their block at the outermost end and applies pressure to the outer as well as the inner wall. The result is compression instead of tension, resulting in less fatigue and the chance to use ceramic liners. There is reduced rather than increased clearance so that leakage is lower in the power stroke and viscous losses are lower on the return.

Swash plate motors induce side loads into their shafts which upset the alignment of the main bearings. Clerk uses spherical bearings.

Every loaded moving surface runs on a hydrostatic pad.

At 1500 rpm and $20 \times 10^6$ N/m$^2$, a unit weighing one ton develops 1.125 megawatts.
GYRO DISCS

Materials

The requirements for energy storage differ from those of precession. The energy people want $I\omega^2$ while the gyro applications need only $I\omega$. This means that, instead of materials with high ratios of strength to density, such as the glass, kevlar or carbon fibre composites, we require a material which has a high product of strength and density. If we want to find the best value for strength times density divided by cost then it is extremely difficult to beat steel. The ordinary mild steels and the low to medium carbon steels with their properties improved by cold working are strong, heavy and cheap. Furthermore, engineers have been using them for so long that we can be confident about their behaviour.

Disc Shape

The newcomer to flywheel technology learns very soon to conceal his surprise that the biggest stresses are at the hub of the flywheel and that even the smallest hole there has a devastating effect on all the stress calculations. Super-efficient energy storing flywheels have obese hubs and knife edge rims and profiles which are cunningly calculated curves.

Again the gyro requirement is different. The most efficient design from the $I\omega$ point of view is a ring wound from piano wire. This offers a very high tensile strength to cost ratio (1400 MN/m$^2$ and £600 per ton, 1979). But although it makes an excellent gyro it is not easy to connect it to the spindle through which large torque must pass and about which large moments must be applied.

Solid forgings have been used for flywheels and give good material properties. They are excellent for small components but, as the weight of the part rises, forging becomes more and more expensive.

The disc design which will be used for duck power take-off is that invented by Robert Clerk (8). It is not the lightest possible because the
gyration forces are so much larger than the gravitational ones that weight is of little consequence. It shape is chosen from the point of view of manufacturing convenience rather than mathematical perfection. It uses very nearly the cheapest steel available (£222 per ton, 1979). It will be operated at speeds which keep the stresses well below the danger level. It needs no machining balancing or even painting. We must forget the image of fantastic precision achieved by navigation gyros and think about plain lumps of steel spinning at a moderate speed.

The disc is assembled from a stack of laminations as shown in Figure 12. Each lamination is pressed into a double cone shape. The lamination thickness is 8 mm, a compromise between the costs of rolling thin sheets and those of pressing the conical shape into thick ones.

The ring nuts which clamp the laminations to the gyro spindle induce in them large precompression forces. These are reduced by the centrifugal forces so that we end up at the working speeds with an acceptable level of tensile stress.

Our present design for a duck of 10 metres diameter and 24 metres width uses four gyros of 17 tonnes each. The total weight of gyro disc is comparable to the weight of pumps saved by the increase of angular velocity.

Applications of the gyro at sea are not new. The first proposal known to me is from Sir Henry Bessemer, better known for his metallurgy. Sir Henry wished to improve the comfort of cross-channel steamers but he was not well advised. He believed that gyroscopic effects only worked over small angles. He planned to use them as roll sensors which would control large rams to stabilise the first class saloon suspended in a rotating gymbal. It is an interesting insight of the class attitudes of the day that this design would have induced larger rolling amplitudes for the second class passengers and crew.

Much sounder grasp of gyro technology is shown by Otto Schlick whose work resulted in the most effective stabilisation of naval vessels.
An MTB displacing 56 tons was stabilised by a gyro disc of 500 kilograms.

Gyro stabilisation was used successfully for over forty ships, the largest being Conte di Savoia of 41,000 tons which used three gyros of 100 tons weight each.

Let me summarise what I see as the advantages of the gyrated power take-off approach:

1. There is a substantial increase of angular velocity.
2. No torque is developed in the back-bone.
3. Oil can be centrifuged to an extraordinarily clean state with no pressure drop.
4. Enough energy storage is provided to run all the later stages at the mean rating rather than at the peak and to let the generators run synchronously.
5. Most important of all, we have got away from the indefinite dangers of sea water and its biology into the best working conditions that the mechanical engineer has ever enjoyed.

RELIABILITY, SIMPLICITY AND MAINTENANCE

The Future Energy Concepts Conference, organised by the IEE in January 1979, contained several papers on wave energy. Some of them contained statements with which I find it difficult to agree.

For example, Glendenning writes (p.112):

A reliable system is one which is simple, contains few, robust components and is readily maintained.

and Bellamy writes (p.168):

The lesson appears to be that the engineering design of wave power devices should provide for large factors of safety with simplicity as the over-riding design principle.
There are no recognised units for the measurement of simplicity, and the various national standards institutions have not suggested approved levels for the engineering profession. But we can turn to the Oxford English Dictionary for a clue to its meanings. There are twelve columns devoted to the concept. They begin harmlessly enough with:

Free from duplicity, dissimulation or guile, innocent and harmless, undesigning, honest, open, straightforward

Free from, devoid of, pride, ostentation or display

(so far so good)   Humble, unpretentious   Free from elaboration or artificiality, artless, unaffected, plain, unadorned   Free from over-refinement, unsophisticated, unspoilt (perhaps unsophisticated is a touch negative? — but worse is to come)   Poor or humble in condition, of low rank or position, undistinguished, mean, common

Ordinary, not further distinguished in office or rank

Not marked by any elegance or grandeur, very plain or homely

Small, insignificant, slight, of little account or value, also weak or feeble (this is all a bit much but we have only covered two columns so far) Low, poor, wretched, pitiful, dismal

Deficient in knowledge or learning, characterised by a certain lack of acuteness or quick apprehension

Lacking in ordinary sense or intelligence, more or less foolish, silly or stupid, mentally deficient, half-witted

With nothing added, mere, pure, bare, single   unlearned, ignorant, easily misled, unsuspecting

The definitions continue with simple heads, simplistic, and so on.

It seems to me that 'simple' is not a simple word. While I do not argue that simplicity is for simpletons, I believe that it is an irrelevant factor. I want to get things right whether rightness comes from simplicity or complexity. The history of technology has many examples of designs which were 'right'. Very often, these 'right' designs are elegant.

The simplest car engine design uses side valves. The most complicated ones use overhead cam shafts. How many side valve engined cars are
Glendenning wants the number of components in a system to be small but does not discuss the manner in which they are arranged. I would argue that the distinction between serial and parallel arrangements is of some importance. With parallel connections, the overall reliability of the system increases with number. There are about $10^{10}$ cells in the human brain. After the age of thirty we lose one per second. Brain surgeons can scoop out teaspoonsful with minimal effect. The brain can be cut into two separate halves and elaborate psychological tests are necessary to detect the operation. I conclude that it is the manner of interconnection of components and their individual suitability which determine success.

The importance of ease of access for maintenance is also frequently urged. For land based plant this is an important feature. It is easier to design things with a short life than for a long one and often more profitable to replace components than to make them last. But if we take the practices of land based plant and try to apply them at sea we have to expect that the maintenance people will have problems getting to work. The more favourable the wave climate the harder they will find it. When I discovered the costs of an offshore man-hour, they seemed so high as to be effectively infinite. When I tried to imagine the quality of work carried out by a man who is cold, sick and frightened, it seemed better to manage without it. Our design group has decided that it should be against our religion to assume that any maintenance will be possible. This concentrates the mind wonderfully and we have tried to apply this principle to every stage of the design. I believe that we can succeed at least as far as the power conversion systems are concerned. Failures will occur and the power canisters must be recovered for inspection so that modifications can be made for future units. The surviving fraction of the population must generate enough electricity to pay for the drop outs.

Bellamy wants large factors of safety. But factors of safety are really factors of ignorance and lead directly to factors of waste. While there must be some band of uncertainty about the magnitude of a force or the strength of a component, there is no reason for this band to be very wide.
We have to balance the cost of reducing the band against the cost of extra, unnecessary material or the costs of failure.

It is also important that we design for the best level of failure. In the field of wave energy this need not necessarily be zero. After thousands of years of evolution the naval architects, ship owners and underwriters have settled on practices which result in between one and two per cent of all ships being lost each year. They have had enough time to get it right. Their factor of safety is less than one.

My own belief is that the correct level of reliability is achieved if:

1. The engineer has an accurate understanding of the working conditions for which he has to design.
2. The engineer has an accurate understanding of the behaviour of the materials and components at his disposal.

GETTING THE POWER FROM THE DUCK TO THE BACK-BONE

The precessing gyros and the ring cam pumps work into a constant pressure manifold and produce a flow which follows the spiky energy flux of the random sea. The pressure is regulated by the variable displacement axial piston motors which pass energy into or out of the gyro flywheels. We have the options of sending power to the back-bone as a constant pressure constant flow of hydraulic oil or as electricity. The oil route would allow the possibility of economically large electrical generators, but we reluctantly rejected this solution in favour of the on-duck generator for the following reasons:

1. For comparable weight of metal, the flow of hydraulic oil in a pipe is less efficient than the flow of electric current in a wire.
2. The flexibility of electric cable is much greater than that of hydraulic hose.
3. We are anxious to prevent cross contamination between hydraulic units.
4. Reciprocating axial piston motors are more efficient and more
easily controlled than turbines but have not been developed for the sizes necessary for exploitation of the big generator option. The on-duck generator can use exactly the same units as are used for the gyro drive and joints.

The generator voltage is a compromise between robust windings and efficient transmission over a distance. The choice of 3.3 KV fits in well with the rule of a kilovolt per mile and a standard range of generator. This sets the current rating of the conductor at a little over 200 amps.

A quick estimate of the flow velocities round the duck in extreme conditions (20 m/sec) reduces the attraction of loose flying leads. We insist that everything should be inside the duck outline.

The cable run must allow angular movements of 270°.

The outline drawing of a duck shows that there is room for a spool of about one quarter of a duck diameter in addition to the main power take-off canister. We propose to wrap several turns of a flat multi-strand belt cable round the spool under and then round to the top of the backbone, as shown in Figure 13. If the ratio of diameters is one to four then the spool will need at least three layers of cable.

When a belt is wrapped around a drum the strain is the ratio of belt thickness to drum diameter. A solid conductor 2.5 mm in diameter would only suffer one thousand microstrain and a seven strand core only 330 microstrain. These strains would allow infinite life for copper or even aluminium conductors. However, our suppliers, W.L. Gore and Associates, prefer to use conductors wound from 37/.4 mm for which the strain will be only $160 \times 10^{-6}$. They propose an insulation thickness of 1.15 mm of CR teflon followed by .4 mm of polyurethane for abrasion resistance. Eighteen of these strands can lie in a pressurised oil cavity inside 2.5 mm of polyurethane. Polyurethane-clad steel strands can be included for strength. An outside layer of polyurethane felt may be added for anti-fouling protection. Fibre optics may also be included to carry data to and from the duck computers.
The CR teflon insulation is extraordinarily effective for its thickness. In air, this cable would be rated for 14 kV AC and 31.5 kV DC, with a 30°C rise at 37.5 amps per core. The belt thickness is 8.64 mm so that the strain in the outer layer is only .35 per cent. This value is extremely small for elastomers.

The rotating spool must be connected to the stationary power canister without slip rings or rotating shaft seals. We can twist it evenly along a 20 metre length parallel to the power canister.

The cable will need a sealed entry to power canister and back-bone. Neither need be mated wet.

When the duck is moving upwards, the torque necessary to accelerate the spool can be provided by belt tension. But it is necessary for the belt to be recoiled on the spool on the return stroke. If ever the torque needed to overcome the spool friction and inertia failed then there would be an ugly tangle of live 3.3 kV conductor.

Model tests suggest that the maximum angular acceleration of the duck is .9 rad/sec². With a 3:1 ratio between duck and spool, the spool acceleration will be 2.7 rad/sec².

A provisional estimate for the moment of inertia of the spool plus the conductor wrapped on it is 450 kg m². This means that we need a torque of about 1200 Nm.

The life time requirement is about $10^8$ operations which makes it difficult to use mechanisms like the torsion bar or the tensator even though we could make them serve double duty as electrical conductors. The cheapest and most reliable mechanism we can devise is a second belt wrapped round the conductor spool as in Figure 14. The extensions of the second belt are small because the amount of conductor wound off the spool will be very nearly the same as the amount of tension belt wound on. The only source of difference is the thickness of the layers. Nevertheless, we should choose a material giving a reasonably low spring rate with a steady tension of about 1000 N. It should have a thin section so that the bending fatigue is kept low.
This tensioned spool design offers the advantages that:

1. The dangerous path between the safety of the power canister and the duck backbone is protected by the hard duck skin.
2. There are no seals or slip rings.
3. The amounts of strain are very small and accurately defined.

MOORINGS

Forces

Every wave energy designer must be familiar with the key equation of mooring developed by Longuet-Higgins and Stewart.\(^{(12)}\) This holds for any device. They showed that, quite apart from the effects of wind or currents, there is a unidirectional leeward force \(F\) determined by the amplitudes \(A\) of the incident, reflected and transmitted waves.

\[
F = \frac{1}{4} \rho g \left[ A_{\text{incident}}^2 + A_{\text{reflected}}^2 - A_{\text{transmitted}}^2 \right]
\]

If we are to minimise the mooring force then it is clear that we must reduce reflections and begin to transmit as soon as the power limit has been reached.

With small to moderate amplitudes, we find that the mean horizontal force on a duck is about 13,000 newtons per metre width for a 10 metre duck. But we do not find that the mean horizontal force rises with larger wave amplitudes. Indeed, the tendency is for the values to fall and some negative values have been measured in extreme irregular waves.

The movement of the duck mounting and the waves created to leeward by large duck angular movements are dumping unwanted energy into the water astern. This is the very best place for it to go.

The negative values of mooring force are caused by the non-linear behaviour of the waves going over the top of the duck. They are broken
up into components with higher frequencies and, as the energy is the same and the period less, the transmitted amplitudes must be higher. This effect is particularly marked with plain circular cylinder models floating nearly awash which can sometimes migrate towards the wave-maker. Longuet-Higgins gives a full account in reference (13).

Practical Moorings

As duck strings present a small profile to long shore currents and the back-bone avoids problems with side-to-side location, our main concern is to resist thrust from the wave direction towards the beach.

We are advised by British Ropes that variation of tension is much more damaging to a rope than a steady large value of tension. Everything must be designed to keep the tension in the cables constant rather than low.

But the safety of the back-bone and duck bearings demands that the duck string must be allowed to yield to any wave with energy more than can be absorbed. If we want lots of movement with the least variation of tension then we must go for a low spring rate.

But low spring rates combined with the inertia of the device will make the natural period of surge movement long, probably several minutes at full scale. This movement can be excited by the envelope of wave groups and, although the exciting forces are low, it is unfortunately the case that the damping at these long periods is also very small. The models go back to the end of their tethers and then come surging forwards towards the waves. The lower we make the spring rate, the further they move, causing great difficulties with electrical connections.

The solution of the problem is to find a way of increasing the value of damping. There are many ways in which the duck string can move. One is surging as a single, unbending body. Another is yawing. The others are separate movements of different sections forming the various modes of vibration. These latter can be damped by the rams in the joints. But another method can be used which will also damp movements of the string as a whole. Damping is force opposing velocity. If we can sense
forward velocities of the string and then modify the duck's power take-off so as to increase the amount of reflected energy then the mean horizontal force will be temporarily increased. When the forward velocity ceases, we revert to the previous low reflection condition. I hope to show that quite small amounts of damping applied in this way will prevent the growth of long period oscillations. Doppler sonar methods will give cheap, reliable velocity indication.

If we design a system of sinking weights and rising buoys which gives a spring rate of 100 N/m² for each metre of duck width and then assume that the ducks follow every wave of the South Uist climate, the mean tension and the variations on the mean are of no concern. British Ropes suggest that we can expect a life of 25 years. They warn, however, that the wires must not be allowed to corrode or bend. Cladding a grease-impregnated cable with a plastic sheath will delay corrosion and we can afford exotic metals to prevent it. The difficult problem is the design of a shackle termination to protect the wires from bending.

To summarise:
1. Never reflect unless you are trying to damp.
2. Transmit when you have absorbed your power limit.
3. Use a low rate mooring.
4. Find a way of damping the long period oscillations.
5. Beware of the terminations.

CONCLUSIONS

Many people have suggested useful methods of device classification in the field of wave energy. Ducks could be described as asymmetric, nearly submerged, surface-piercing, deep water, low-Q resonant, crest-spanning, hard-skinned, close-packed, end-stop free, capsizable, overload shedding, rotary terminators on active mountings. Their maximum swept volume is about twice their displacement. Their mean power limit is 7.5 kilowatts per square metre of projected area, and about .6 kilowatts per ton. The high pressure oil hydraulic power take-off is hermetically sealed, energy
storing, designed for zero maintenance and electronic control. Generation is synchronous parallel connected AC with surface accumulation. Ducks use bulk high voltage AC transmission.

ACKNOWLEDGMENTS

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DUCK EFFICIENCY on COMPLIANT AXIS
monochromatic sea: $T=12\text{sec}; L/d=22$

data rescaled to represent 10m dia duck

FIG. 1
Fig. 4

Silverline Hydrodynamic Bearing

Fig. 5

Slubber with valve and bellows
APPENDIX 9

SPINE JOINT ANGLE PROBABILITIES
Hydrostatic power transmission has been disadvantaged until now by unacceptable noise, high relative cost, poor part-load efficiencies, low power/weight ratio, and bulky control ancillaries. Eight years of fundamental research and composite development covering fourteen problem areas primarily to achieve characteristics complementary to an energy storage flywheel have resulted in a hydraulic pump and identical motor of unusual capability, performance, ease of manufacture, and above all, controllability by virtue of a modular, programmable, hydro-computing element, inserted within the pump/motor casing, and replacing all the usual control impediments, excepting the operators command transmitter. Extremely low zero-load running losses, in continuous 200% overspeed operation, advantage applicability to flywheel energy systems as does its reciprocal capability as a motor which will accept hydraulic power and transmute it into rotary flywheel energy or which will serve as vehicle wheel-driving motor offering real time individual wheelslip control whether accelerating or being used as a hydraulic retarder. It can operate entirely without bulky and costly ancillary equipments such as pressure relief valves, filters, heat exchangers and system control valves. Its power rating can be stretched almost 100%, its operating speed range 300% and its life expectancy 1200% without production change other than materials specification. Yet perhaps its most outstanding characteristic is its simple smoothly modulated instantly responsive control.

PREFACE
In another paper presented in this symposium, we have discussed a high energy accumulator in which a storage flywheel is associated with a specially developed pump. But although the accumulator is a complete product in its own right, it is only half the story in any application of flywheel energy to a vehicle or other mechanism which might be powered by rotary motor output. This paper describes development of the special characteristics necessary to achieve functioning and performance complementary to storage flywheel operation and to application in a vehicle as automatic power transmission which with the power unit or units and controls form an integrated propulsion system.

THE COMMISSIONAL BRIEF
As no hydraulic power pump existed, or was in prospect, which could in any way operate in function complementary to a storage flywheel, a specification was drawn up of indispensable and expedient characteristics which appeared feasible and could serve as a commissional brief.

What was required was a vertical axis variable pump having the following characteristics:
(1) consistent operation within the 20C% overspeed range above rated power speed at rated power or less.
(2) ultra-low rotation drag loss in zero-load running at 200% overspeed or less.
(3) mechanically and hydraulically quiet running and pulseless intake and delivery.
(4) high mechanical and hydraulic power transmutation efficiencies whether pumping or motoring.
(5) as a motor, displacement must go over-center for reverse driving.
(6) wholly internal all-hydraulic control system with data processing capability which will operate in harmony with other similarly equipped elements of a complex, as an integrated system exercising perfect
control modulation and response. (7) optimum commonality between pumps, motors and pump/motors, vertical or horizontal axes, and right or left handed rotation.

(8) high power/weight and power/bulk ratio.

(9) low cost per horsepower rated throughput or per torque unit.

(10) minimal or zero dependence on external ancillaries; e.g. circulation filters, pressure relief valves, oil heat exchangers, actuators, etc.

(11) simple installation; all hydraulic connections into manifold facing on pump (top) end opposite driving end. Drivehaft to be only mechanical connection. Smooth exterior devoid of screws and flanges.

(12) pump/motor to be stretchable in power rating and/or life expectancy merely by changing materials and treatments of basic design.

(13) eschewing empirical factors, design principles to be based on logical variation of accepted engineering practices, so reducing development costs of a range or family of pumps and motors.

Unfortunately no established hydraulics manufacturer would accept what they considered to be an “impossible” commission, so it landed back in our own project office.

HISTORICAL RESEARCH

Unfortunately one must pass over the capable work of Thomas and Hale-Shaw and go back to Williams-Janney, 1907, for logical precepts applied to a ported axial pump, instead of a melange of inconsistent empirical factors.

Thomas’ fixed displacement bent-axis pump is a real gem, but the variable displacement development is a real Rube, but no ruby. The Williams-Janney variable design, with no notable improvement, is still being manufactured for marine application, offering a dependability and reliability not found in modern slipper-pad pumps.

MARKET RESEARCH

Market research showed there was real need, outside of the low power bracket, for a pump embodying many of the characteristics in our commissional brief, but more importantly, automotive hydrostatic transmission was awaiting the viability of just such characteristics.

This provided our first go-ahead decision point.

FEASIBILITY STUDY

This pinpointed the problem areas in which normal pump design was deficient, and the study areas for which classic engineering and commercial solutions had to be found. These were:

1) Compatible working fluids
2) Filtration - necessity or fetish?
3) Internal hydrodynamic “churning” losses
4) Manufacturing precisions and operating strain distortions. Rigidity or conformability
5) Transmission of driving torques from or to the mainshaft.
6) Transmission of driving thrust to and from the pistons.
7) Cylinder barrel: the costly heart of a pump.
8) Curbing fluid-shear losses at high rotational speeds.
9) Portface fluid-shear, flutter and port area control by clearance modulation.
10) Port-flow compatibilities.
11) Hydraulic decompression shock and mechanical noise.
12) Flow ripple and pressure pulsation.
13) Hydraulic data processing and actuating controls
14) Integration of problem solutions into a commercially acceptable composite adaptable to both high and low volume production techniques.
15) Elemental commonality and integration of complex systems.
16) Materials technology as affecting life expectancy and cost effectiveness.

RESEARCHED SOLUTIONS

Each of the above areas was exhaustively studied and the possible solutions short listed to those most suitable to production processes and communal integration.

COMPATIBLE WORKING FLUIDS

Hydrocarbon hydraulic oils are like a sponge, absorbing upwards of 8% air into solution at standard temperature and pressure, more at higher pressures, and will release this as discrete bubbles at any sudden pressure drop within the pump circulating system. This leads to premature cavitation and subsequent pressure hammer, both of which have destructive effect.

The working fluid should retain minimal discrete or dissolved gases and, needless to say, must also be non-corrosive and a reasonable lubricant; such as Losorb Silicone for atmospheric application or vacuum-stripped
Hydrocarbon for evacuated applications. An hermetic reservoir, circuits and sealed systems would be a distinct advantage. Surface dissipation should keep the reduced waste heat with 45°C above ambient.

Filtration

Axial piston pumps are notoriously sensitive to anything less than perfect filtration - some so sensitive that just breathing the word "grit" will cause them a seizure. I just am not a believer in hydraulic circuit full-flow filtration - suction or pressurized. The one clogs, unless regularly serviced, with results which can be more disastrous than unfiltered flow; the other is too heavy, bulky and costly, yet never 100% safe.

The aim should be to design a pump or motor which will digest anything which will negotiate the induction and delivery passages and galleries without physically jamming up the works. The advanced pump under discussion must be so designed that by a change of material specification any (lack of) filtration requirement can be met. Sensitive flow impedences and bearing surfaces should be individually guarded by self-clearing porous elements.

Internal Churning Losses

All pumps run with their casings full of oil (at least until my earlier published work seven years ago) in order to damp mechanical and hydraulic noises and to ensure adequate lubrication. Any internal ullage results in severe aeration and real trouble, but even with zero ullage hydrodynamic "churning" creates a resistance drag and heat problem.

The only answer, for a pump particularly, is to go "dry sump" and scavenge internal leakage oil as quickly as possible; or go further and evacuate the air/oil mist as well. At the speeds entertained for the energy storage application, oil churning would waste literally hundreds of horsepower as embarrassing heat. As regards the undamped mechanical noises, these must be eradicated at source.

Precision: Rigidity vs. Conformability

Pump design is normally based on very precise relationships within a rigid structure. Unfortunately rigidity can never be absolute, depending upon the applied load relative to mass and modulus. Precision also is relative, and the closer it approaches absolute the more rapidly do manufacturing costs escalate according to inverse cube law.

By adopting the principle of flexible or articulated component inter-relationships in association with conformable structuring, the precision requirement and also the mass modulus (weight) requirement can both be reduced by an order of magnitude, particularly if the ancillary equipment items are designed as insertable modular sub-assemblies.

As example, if all the components connecting the upper casing assembly to the lower case assembly are articulated (e.g. ball ended connecting rods) this seeming extravagance will allow very thin and flexible casing skirts which if compressively prestressed by end-to-end elastic tie-rods will be audio-frequency stiffened and otherwise competent so long as the displacement tilt-box is not gimbal-journalled in the casing skirt.

This applies to all component inter-relationships except the portface which in any event is at the interface between the only two deep sectioned masses, the cylinder block and the casing head. The cylinder sleeves float in the block, the short pistons in the sleeves, the cross-axial rods in the pistons and to the swash-plate which floats on rockers in a rocker journal which floats in the lower half casing. The mainshaft also is spherically journaled in upper and lower casings and employs a floating link drive to the swash plate, so is free to bend or run misaligned without ill effect. As a result no spigot or dowel location or bolting flange is required at the inter-casing joint face, and casing skirt sectional thickness is foundry determined.

Transmission of Driving Torque

In most axial pumps the drive from the mainshaft is through the attached cylinder barrel via the pistons, the cantilevered ends of which carry articulated slipper-pads which apply the load to an angled swash-plate. Load reaction and friction, of that part of the piston remaining in the cylinder, is enormous. At high loads and swash angles the leverage-fractions and the bending of the pistons provide a definite limitation to torque and horsepower transmitted. What is more the cylinder barrel
is tipped over at the port-face causing wear at the contact edge and pressure leakage at the open edge.

Driving direct from the mainshaft to a tiltable rotating swash-plate overcomes all these deficiencies. Williams-Janney did it with double-Hooke's 3 imbal with limited angle of tilt, Hele-Shaw with spherical ball universal with torque Brinelling limitation. We opt for a spider-link universal Fig. 1 and Fig. 2, allowing not only high torque and large tilt angle but tying the swash-plate radially to the mainshaft yet allowing it free axial float and tilt.

But the piston pressure loads, acting on the tilted swash-plate, generate a combined lateral component effective as torque at a mean 0.707 of active radius, whereas the entire lateral load must be located, in a slipper-pad pump by the shaft via the cylinder barrel and the extended pistons, in our case directly by the mainshaft which can bend to the strain without affecting the cylinder barrel in which is rotated in synchro-nism by three resilient fingers.

At large angles of tilt, the geometry corrections to improve compliance and life of the spherical link joints cause a slight epistrochoidal variation from true concentric rotation of the swashplate, but its only effect is the inertia harmonic transmitted by the "tight" links to the mainshaft.

The calculated efficiency loss of this universal drive was 0.75%; proved in tests as 0.86% using off-the-shelf Unibal joints instead of the intended Tungsten carbide/cast iron(molybdenum flashed) joints. These are a press/adhesive fit in the light alloy swashplate and in the cast lugs of the modular iron mainshaft, the drive pin through each joint ball being connected by a parallel pair of strain-compensating links to the complementary joint pin as a closed loop.

TRANSMISSION OF DRIVE THRUST

As there is no torque transmission between cylinder block and swashplate, the pistons can be short light-alloy die-castings Fig. 3 thrusting via ball ended connecting rods to the rotative swash-plate.

---

Swashplate
Counterweight
Shaft
Open-C drive link (3)

Fig. 1 Universal Spider Drive: open links.

Fig. 2 Universal Spider Drive: Symmetrical Closed Links
The piston/con-rod compliance never exceeds 2\(^\circ\) whereas the con-rod/swashplate compliance approximates to the tilt angle and is accommodated by standard Glacier DX cup and truncated cap lubricated and semi-floating by cylinder pressure transmitted via the drilled connecting rod.

The angle-contact paralleled thrust bearings shown in Fig. 2 have given way to an all-hydrostatic thrust pad arrangement which has the advantage of "floating" any tilt-box distortion and of being retractile to reduce shear-drag at low piston thrust loads. Thrust compliance and support efficiency losses amount to 0.65\% using Carbon steel connecting rods, unplated.

The tilt-box is supported on a pair of rocker journals which are hydrostatically floated in a turret assembly Fig. 4 itself floated in the lower half casing so that it can be rotated the few degrees necessary for port-face control of decompression energy as later discussed. Tilt-box rocking actuation for displacement control is effected by a parallel pair of cylinder/piston/con-rod assemblies identical to the pumping assemblies but having the cylinders inserted in the upper casing.

The "as-cast" cylinder barrel is assembled to the port-plate with the O-ring floated liners entrapped within the rough bores, and is located and secured by a circlip and a filler adhesive. The only metal removal would be on the dynamic balancing rig.

Cylinder Barrel

Normally the most expensive part of a pump, and having the highest reject percent scrap, whether the cylinders are honed in situ or pressed-in liners. The repeated hydraulic pressures and shock waves induce hoop strains which can lead to crazing and break up of the bore surface.

We have developed a fully-floating wet liner Fig. 5 which, being subject to the same pressures outside as inside, is always in some degree of compression. This allows a wide choice of tensile-sensitive fabricated cylinder liners, appropriate to life and application, e.g. ceramic or iron sintered, cast white irons, or low cost cold-extruded finished steel tubular sleeves.
Apart from the production and life advantages, the pressure floated sleeves are perhaps more important operationally, as they are not subject to hoop strain expansion by the pressures within the cylinder and therefore to the clearance cubed leakage law. On the contrary they can be designed to close up under pressure so that leakage is neither pressure related nor a function of piston length.

Although the high-alumina ceramic sleeves, in association with plasma-sprayed hard coat pistons, were intended primarily for infinite life applications, their inertness to fire resistant or non-flam hydraulic fluids is opening up new avenues of application activity.

The 6° radial splay of the cylinder is a convenient way of accommodating the optimum portface diameter to the most effective swashplate socket-circle diameter for the connecting rods, but it has minor advantages in that the centrifugal component of the splay angle reduces piston pull-out forces and assists the fluid flow into the cylinders during the intake stroke.

CURRING FLUID SHEAR LOSSES

Having come to the conclusion that ball bearings were too noisy and put an effective limit on pump performance and therefore on rating "stretch", also that roller bearing shaftness noisy were at a disadvantage in overspeed running, it became necessary to find ways of reducing the film-shear drag of fluid film bearings and raising the frequency response without affecting the excellent noise damping characteristic.

Hydrodynamic bearings were found unsuitable in several areas, so hydrostatic bearings were chosen, and researched in great depth to minimise both shear and circulating eddy losses and to reduce the pressure-leakage power. Hydrostatic journal bearings posed a greater problem than thrust bearings and so were treated separately, but the biggest problem was the portface which being rateless would not respond to any direct application of the hydrostatic art.

Portface Shear Drag This is a function of shearing surface area of the sealing lands, the mean operating radius, differential surface speed, and the running clearance. Having optimised the portface dimensions, and being required to maintain the wide overspeed range, we could only work on the running clearance.

Portface Clearance Control At high pumping pressures, prevention of port-face leakage demand a close precise running clearance, allowing only sufficient leakage across the lands to flush away the shear heat—quite considerable for a largish pump running at really high revolutions. At lower pressures the same clearance would not allow enough flushing leakage to prevent a hot spot or worse, when running "free". We therefore contrived a floating counter-thrust arrangement Fig. 6 whereby the clearance varies inversely with pressure and in fact at very low pressures or "free" running, it opens a bit extra to reduce the shear loss to negligible proportions.

In any normal pump, the port sizes are very definitely limited by the hydraulic forces holding the port-face in balance. The contrived counter-thrust provides sufficient control over these forces to allow doubling or trebling the port areas as may be necessary to prevent induction cavitation at the flow rates consonant with a large power throughput.
Piston/cylinder Shear Drag

Discussed earlier, the floating ambience cylinder sleeve allows an inversion of normal practice, in that we can start with a generous sliding clearance which closes up with pressure to reduce leakage and shear drag in a controlled manner.

Hydrostatic Thrust Bearings

Primarily the portface counter-thrust and the swashplate thrust support bearing, which posed different problems. The former being a continuous annular arrangement was difficult in that it demanded 30 k Hertz response with minimal leakage from the working pressure bleed supply and was achieved by triple phased impedances with the float pressure taken after the first stage. This was so successful that a similar arrangement was adopted for the swashplate support bearing though initially this is fragmented into 10 floating circular pads, five on the intake semicircle and five on the delivery side, three on each side being retractable at low pressures to reduce no-load shear drag; but these will later be replaced by a pair of semicircular pads having lower overall drag.

Hydrostatic Journal Bearings

Primarily the portplate sleeve-shaft juxtaposed journal pair, and the mainshaft spherical main journal bearings, the shear drag of the former has been reduced mainly by tailoring the land running clearances inversely as and the land widths directly as the maximum operating pressure drop across the lands. The spherical main bearing Fig. 7 has been similarly treated but the greater diametral surface speed generated severe eddy current losses now controlled by chevron bar lands and recirculating perimeter channels. Each bearing has only three unequal pad areas as the loadings are unidirectional.

PORT-FLOW COMPATIBILITIES

Because the pistons induce axially, most other pump port galleries have axial entry. Unfortunately, in any pump and particularly in a high speed pump, the passing speed of the cylinder ports around the port-face Fig 8 can be a whole order of magnitude greater than the axial velocities, resulting in spill-wave formations which deflect, and can effectively choke, the axial flows into the following cylinder, causing disastrous cavitations.

We have designed the portface with tangential inflows accelerated by a convergent gallery to the mean passing speed of the cylinder ports. Fig. 9

![Fig. 9. Velocity Match at Fixed Displacement and Convergence.](image)

However, to cater for wide variation of flow at constant speed, or wide ranges of speed at constant flow, even this is not enough. It has been necessary to develop a portface with internal supercirculation, forced by the passing cylinder ports, and

![Fig. 7. Spherical Hydrostatic Main Journal Bearing.](image)
amplified by a controlled vortex. Fig. 10 shows this for the overspeed case with positive supercirculation "gain".

**Fig. 10:** Velocity Match by Supercirculation at Low Angles

\[
\Delta V = \text{Venturi-Accelerated Flow Factor} \\
S = \text{Supercirculation Gain (Auto-Variable)} \quad \text{where } V_2 > \Delta V, S
\]

and Fig. 11 for under speed "negative gain".

**Fig. 11:** Velocity Match by Retarded Circulation at High Displacement Angles

\[
\Delta V = \text{Retarded Circulation Factor (Auto Variable)}
\]

The internal profile has a self-polishing fluidized-bed deposited finish.

**Radial Splay of Cylinder Port-tracts**

Even with the large cylinder-ports, the acceleration of the oil into and through the port-tracts might need to be greater than can be effected by the residual absolute intake pressure. Splaying of the port-tracts provides a centrifugal component acting to accelerate the oil column in the port-tract, aided by the centrifugal pressure generated by the supercircuiting flow in the arcuate port-face gallery.

**DECOMPRESSION SHOCK AND NOISE:**

When each piston completes its pumping stroke at top-dead-centre, and its cylinder port passes from the pressure port-face, across the sealing land to the intake port-face, the residual pocket of high pressure oil is released into the intake with a resounding shock-front which is difficult to control at differing flow rates and passing speeds. Conversely, when passing across bottom dead centre the cylinder full of low pressure intake oil is suddenly in communication with the pressurised port-face, and an even greater shock-wave will enter the cylinder, travel down it and hammer the piston and cylinder walls. This is the reason for the peculiar shape of the piston crown Fig. 3 adopted as a result of experience of high-energy-rate explosive forming.

A simple but effective way of dissipating this pent-up pressure energy is to "bleed" the dead-centre sealing land via a porous impedance which will allow fluid to pass, but not a shock-wave, so that there is no pressure differential when the cylinder port reaches the next port-face area. Unfortunately the pent energy is wasted by this ploy.

A more efficient arrangement is to adjust the port-face timing so that opening to the next port-face area is delayed beyond top dead centre until the piston has retreated (been forced back) until the compressed oil has expanded to intake pressure, or conversely delayed beyond bottom dead centre until the piston has advanced far enough to pre-compress the cylinderful of fluid to delivery pressure.

This has long been recognised as difficult of achievement, but is now effected quite simply by rotating the tilt-box rocker turret to the appropriate lag angle, automatically, manually or by preadjustment dependent upon application.
FLOW RIPPLE AND PRESSURE PULSATION

Every piston pump has a volumetric variation, dependent upon the number of cylinders and their displacement.

The result of this flow ripple in a "hard" hydraulic circuit (as distinct from a spongy one) is a pressure pulsation the amplitude of which increases with "hardness" to destructive proportions. Adding a small air/oil accumulator to the circuit will serve as a palliative up to choking frequency, a line-damper "accumulator" will raise the safe frequency of the closed system as a whole, though less so at the source point.

For frequencies of the order of 1500 pulse Hertz (9 cylinders at 10,000 RPM) it becomes obligatory to control the pulsations substantially at source point. Even less appreciated is the necessity to control the induced pulsations in the intake port as even a low amplitude can be the source point of suction cavitations. Fig. 12 shows the solution to the problem, incorporated in our new pump design, a modular air/oil pulse absorber, inserted into a cavity galleryed with one port or the other and precharged to two thirds of the limiting pressure above and below ambient respectively.

HYDRAULIC DATA COMPUTING CONTROL

Where one or more variable pumps is operating in association with one or more variable hydraulic motors, the interrelationships can be quite complex, even for a simple duty cycle.

For complex operations the control system can be nightmare, like a well known hydrostatic transmission for light tractors, where the pump and motor are lost within a welter of links, levers, cams, cranks, servos and other appendages.

We have developed a small computing valve module, Fig. 13 which is inserted into a cast pocket within the pump/motor unit, and serves to prescribe all the operational functions in relation to like-programmed computing valves inserted in the pumps or motors elsewhere in the system.

Except for the operators (driver's) pedal and lever control inputs Fig. 14 (which provide fluid pilot signals), there are NO mechanical appendages or interconnections in the system, only the minimum of data or pilot-line connections between the pre-designated master valve and the slaves. In fact, where there is only one slave unit and therefore no ambiguity this minimum can be zero - NONE.
This computing valve, with up to 8 input/outputs, can handle quite complex operational requirements. The most complex application so far considered is for a kinetic accumulator boosted vehicle transmission, where the master valve in the accumulator pump resolves and controls all but two of the 19 operational requirement functions, even including the mass energy summation equation; $K = m \ddot{V} + I (\alpha_2 - \alpha_1)$ (constant) in the regenerative braking and charging modes. Except in "Neutral", it controls the vehicle engine absolutely, ensuring optimum b.m.e.p. in all regimes, and it provides instantaneous modulated response to operator demand, regardless of engine response characteristic. All this and much, much more, with a charming simplicity.

COMPOSITE DESIGN INTEGRATION

As each problem solution had been selected on a basis of interrelating compatibility, their integration into a composite design Fig. 15 did not prove difficult.
FIG. 15A

Composite Arrangement.
Of course they would not necessarily all be incorporated together, but it would perhaps be cost effective for the same production arrangement to allow this.

For example, the supercirculating intake gallery Fig. 16 has special relationships to flywheel systems, but equally a pump so equipped could be driven from a higher ancillary drive gear stage of a gas turbine; or directly by a high speed gasoline engine, merely by programming the hydro-computing insert to the appropriate speed and delivery parameters. Or the pulsation controllers may be an application luxury and the cavities plugged.

Excepting the port galleries in the upper casing all casting dies (or alternative sand patterns) can be straight draw and will accommodate alternative materials despite differing shrinkage, working stresses being calculated to the lowest factor. The lowest cost cylinder sleeves are from cold-extruded carbon steel tube, with bores unmachined but ballized, and cost a few cents each, whereas at the other extreme high-alumina sleeves cost a few dollars each, but well worth it for some applications. The lowest cost pistons are pressure die cast and pushed through a plain broach before anodizing, those at the other extreme are plasma sprayed complementary to the cylinder sleeves.

**COMMONALITY IN COMPLEX SYSTEMS**

Pumps and motors are identical except for the inserted hydrocomputing element which determines their characteristics and except for the priming mini-pump which is not always necessary. Handling, where necessary, if effected when machining the portface and by fitting a handed priming pump if required.

Every unit, pump or motor, generates a pressure signal varying the square of speed and this provides an element of data for computer processing. Another data element can signal displacement tilt angle. Modulation of return line pressure will provide further data which together with a common basic response to the system operating pressure provides identical-twin or family interrelationship between any number of units programmed to similar basics. Response can be immediate and sharp or allowed a fixed or proportioned time rise.

**APPLYING MATERIALS TECHNOLOGY**

Although this pump or motor is designed for low cost production in high or low volume, and the basically rated unit utilises low cost material for limited life requirement, we have established that it is generally more cost-effective to increase rated throughput and life by substituting better but more expensive materials treatments and finishes within the same design dimensions and on the same production and assembly tools and fixtures than to have to design develop, tool up and provide additional spares holdings for a larger size of pump or a more closely spaced range of pumps.

The basic "Lo" rated pump can be "stretched" to a "Normal" rating 40% higher, or further to "Hi" rating 80% to 90% higher; alternatively its operating overspeed range can be "stretched" to 200% higher, all with appropriate life ratings, by change of materials specification. For example, connecting rods can be carbon steel, plated Aluminium or Magnesium alloys, or Titanium alloy, depending upon operating pressures and overspeeds. Cylinder sleeve and piston alternatives have already been mentioned; as have casing materials which can be cast iron for underground application, high-damping bronze for undersea,
Aluminium for normal above ground use, or Magnesium for ultra-quiet or ultra-light special requirements, all from the same patterns or dies without shrinkage corrections. The port-tract supercircular centerbody which complicates castings is being developed as an optional insert but we are researching a controlled vortex without centerbody, also a precast port-face and tract burnt into the light alloy casting.

MODERNISED ASSEMBLY METHODS

In this design, production screw threading has been generally tabooed, and fixings are by elastic tierods, self-loading bevel circlips and spring circlips, and interference or adhesive filled permanent fits: these may even be used in combination to provide "belt and braces" security; the adhesive located piston collets have a back-up wire circlip, and bevel circlip for a high frequency location has an adhesive security bond.

EXEMPLARY PUMP/MOTOR PERFORMANCE

A 25 mm bore 9 cylinder axial piston pump (or motor) having a displacement variable up to 16.8 cu. in. per revolution, scales 152 lbs with all ancillaries built in, and measures 10¼" x 11" x 18" overall excluding the driveshaft projection. In "Lo" rating it is controlled to a maximum delivery of 162 GPM (US) and 4400 p.s.i. (416HP) at 2500 RPM but may be programmed to give a higher pressure at commencement of delivery to a motor, allowing greater starting torque.

In "Normal" rating it is controlled to 195 GPM maximum and 5140 p.s.i. (570HP) at 2800 RPM and in "Hi" rating will deliver 216 GPM at 6100 p.s.i. and 3000 RPM or above. When being motored with zero delivery, the driving torque is only 1.03 lb. ft. at 2500 RPM and 1.43 lb. ft. at 3000 RPM rising to 4.2 lb. at 9000 RPM with heavy "Lo" rated internals at present state of development. Leakage (including lubrication) is 2.48 GPM at 4400 p.s.i. falling to 0.6 GPM at 1000 p.s.i. below which it remains constant. Mechanical efficiency of approximately 98% is suspect until our new back-to-back dynamaneter is commissioned. Noise is entirely related to port timing adjustment which is not yet in automatic mode.

The pump features are protected by ten U.S. patents and the initial production range is exemplified in Table A.

<table>
<thead>
<tr>
<th>Coded Rating</th>
<th>Dimensions (in. (B x W x L)</th>
<th>Weight lbs &amp; Mattl.</th>
<th>Displacement (cyl/comp)</th>
<th>Pressure (Op.psi)</th>
<th>Torque 100% R.P.M. (lbf. ft.)</th>
<th>0.25 (Rated) R.P.M. (lbf. ft.)</th>
<th>H.P. (Rated H.P.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>7/12/20/Lo</td>
<td>41/8 x 41/8 x 71/8</td>
<td>10 Al.</td>
<td>1.22/20</td>
<td>5140</td>
<td>83</td>
<td>6000</td>
<td>31</td>
</tr>
<tr>
<td>7/12/27/Lo</td>
<td>51/2 x 51/2 x 81/2</td>
<td>32 C.l.</td>
<td>1.65/27</td>
<td>4400</td>
<td>96</td>
<td>5250</td>
<td>37</td>
</tr>
<tr>
<td>7/12/30/Norm.</td>
<td>51/2 x 51/2 x 81/2</td>
<td>17 Al.</td>
<td>1.83/30</td>
<td>5140</td>
<td>124</td>
<td>5750</td>
<td>65</td>
</tr>
<tr>
<td>7/12/33/H1</td>
<td>51/2 x 51/2 x 81/2</td>
<td>12 Mg.</td>
<td>2.01/33</td>
<td>6100</td>
<td>162</td>
<td>6250</td>
<td>54</td>
</tr>
<tr>
<td>7/15/60/Norm.</td>
<td>61/4 x 61/4 x 101/4</td>
<td>33 Al.</td>
<td>3.66/60</td>
<td>5140</td>
<td>269</td>
<td>1600</td>
<td>72</td>
</tr>
<tr>
<td>7/18/100/Norm.</td>
<td>71/8 x 81/8 x 131/8</td>
<td>50 Al.</td>
<td>6.1/100</td>
<td>5140</td>
<td>415</td>
<td>4000</td>
<td>103</td>
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<tr>
<td>7/21/155/Norm.</td>
<td>81/4 x 91/4 x 151/4</td>
<td>90 Al.</td>
<td>9.5/155</td>
<td>5140</td>
<td>647</td>
<td>3300</td>
<td>133</td>
</tr>
<tr>
<td>7/25/250/Lo</td>
<td>101/4 x 111/4 x 181/8</td>
<td>280 C.l.</td>
<td>15.25/250</td>
<td>4400</td>
<td>888</td>
<td>2500</td>
<td>162</td>
</tr>
<tr>
<td>7/25/265/Lo</td>
<td>101/4 x 111/4 x 181/8</td>
<td>152 Al.</td>
<td>16.35/265</td>
<td>5140</td>
<td>1115</td>
<td>2800</td>
<td>195</td>
</tr>
<tr>
<td>7/25/280/81</td>
<td>101/4 x 111/4 x 181/8</td>
<td>99 Mg.</td>
<td>17.0/280</td>
<td>6100</td>
<td>1375</td>
<td>3000</td>
<td>216</td>
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<tr>
<td>7/35/320/Norm.</td>
<td>151/2 x 151/2 x 251/4</td>
<td>150 Al.</td>
<td>50/820</td>
<td>5140</td>
<td>3600</td>
<td>2000</td>
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<tr>
<td>7/50/230/Norm.</td>
<td>211/2 x 211/2 x 361/2</td>
<td>1215 Al.</td>
<td>130/2310</td>
<td>5140</td>
<td>8850</td>
<td>1500</td>
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<tr>
<td>7/60/380/Norm.</td>
<td>251/4 x 261/4 x 431/4</td>
<td>1215 Al.</td>
<td>232/3800</td>
<td>5140</td>
<td>15800</td>
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<tr>
<td>7/72/660/Norm.</td>
<td>311/8 x 311/8 x 521/4</td>
<td>3620 Al.</td>
<td>271/6600</td>
<td>5140</td>
<td>27140</td>
<td>1000</td>
<td>1709</td>
</tr>
<tr>
<td>9/100/17200/Norm.</td>
<td>41/2 x 141/2 x 701/2</td>
<td>9700 Al.</td>
<td>1050/17200</td>
<td>5140</td>
<td>71500</td>
<td>750</td>
<td>3360</td>
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