A hydraulic power-take-off was designed and built for outboard engines typical of those used on small boats and Pacific City-type dories. An associated hydraulic system was developed for use on these boats to provide power for fishing machinery. The power for each system was provided by a hydraulic pump direct-drive coupled to the outboard engine crankshaft at the flywheel.

Four 1970 model outboard engines were used during the project for the design and testing of these systems. These engines were: a 40 hp and a 60 hp Johnson, a 50 hp Mercury, and a 55 hp Fisher-Pierce Bearcat.

A concurrent effort involved the design and development of light-weight hydraulically-powered salmon gurdies especially configured for use on small boats and dories.

Four Pacific City-type dories, each equipped with one of the above engines and its associated hydraulic system, were tested
throughout the 1970 summer salmon commercial fishing season on the Oregon Coast. These boats logged a total of 2,725 hours of operating time during the season and accumulated gross earning of approximately $27,200. This is an average production of $8.50 per operating hour versus an average production of $5.50 per operating hour for such boats equipped with hand-powered gurdies.

Subsequent analysis of operational data, engines, and hydraulic system components support the conclusion that a reliable, efficient, and relatively inexpensive hydraulic power-take-off system for outboard engines to power on-board fishing machinery is definitely practical.
Design and Development of a Hydraulic System for Outboard-Engine-Powered-Commercial-Fishing Boats

by

Charles Alan Marshall

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Typed by Opal Grossnicklaus for Charles A. Marshall
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operational testing of the systems; and Donald Swantes, Newport, Oregon, outboard mechanic and fisherman, for his advice and mechanical assistance.

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NOMENCLATURE

T  Torque in inch-pounds
P  Line tension in pounds of force
R  Radius of spool or reel in inches
S  Spool or reel rotational speed in revolutions per minute
L  Lineal line speed in inches per minute
π  Dimensionless ratio equal to 3.14
RPM Revolutions per minute
HP  Power in units of horsepower
HP_p  Power required by a hydraulic pump
η  Hydraulic system efficiency factor
Q_G  Heat generated in British thermal units (BTU) per hour
Q_C  Heat dissipated by convective heat transfer in BTU's per hour (BTU/hr)
ΔT  Temperature difference in degrees Fahrenheit (°F)
\( \bar{h}_c \) Average convective heat transfer coefficient in BTU per hr per square foot per degree Fahrenheit (BTU/hr-ft² °F)
## DEFINITIONS

<table>
<thead>
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<th>Term</th>
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<tr>
<td>Pacific City Dory</td>
<td>An 18 to 22 foot long wooden boat with high upswept bow for surf launching; generally powered by an outboard engine mounted in an inboard well, rated at from 30 to 80 horsepower; indigenous to the Pacific City area of the Oregon Coast.</td>
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<td>Gurdy</td>
<td>A small winch, powered or hand driven, used in the commercial fishing industry to haul in fishing lines and other gear.</td>
</tr>
<tr>
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DESIGN AND DEVELOPMENT OF A HYDRAULIC SYSTEM FOR OUTBOARD-ENGINE-POWERED-COMMERCIAL-FISHING BOATS

I. INTRODUCTION

Background Information

Outboard-engine-powered boats are an important part of the equipment of the total commercial fishing industry in the United States and other areas of the world. The gillnet salmon fishery in Alaska, Pacific Northwest trollers, bay scallop dredgers in New England, in-shore hook and line, and pot fisheries in the southeastern United States are a few examples of fisheries where outboard-engine-powered boats can be used to advantage because of their high speed, relatively low cost, and ease of maintenance.

The inherent economic advantages of this type propulsion for certain fisheries has been limited by the inability to secure a power-take-off system which can propel simple fishing machinery such as troll salmon gurdies, line haulers for longline trawl and pot fishing, line pullers for surface trolling, or gillnet reels.

Fishing capability and productivity of outboard-engine-propelled boats can be greatly improved if the fishing gear can be operated by adequate mechanical power rather than manpower. However, the development of such a system powered by an outboard engine must
meet certain requirements. To be efficient, the system must:

1. Conserve weight and space; as outboard-powered fishing boats are open skiffs and/or dories where extra weight or sacrificed space would limit fishing efficiency.

2. Be simple, relatively inexpensive, and resistant to the corrosive effects of salt water.

3. Not affect the operating cycles or life of the outboard engine.

4. Assure the fisherman a power supply that can be used for a variety of fishing operations and fishing machinery without radical or expensive changes of power take-off equipment.

5. Deliver adequate fishing machinery propulsion power over a long time span and offer easy maintenance.

Larger fishing boats in the 30 foot plus class are propelled by automotive-type inboard engines and the onboard fishing machinery is normally powered by mechanical, electrical, pneumatic, or hydraulic systems. Hydraulic systems, because of their versatility, easy maintenance, minimal space requirements and high efficiencies with consequent lower power requirements from the propulsion unit have certain distinct advantages over the other systems in small fishing boats.
Objective of Project

The primary object of this fishing gear project was to apply the concepts of hydraulic power transmission to boats powered by outboard engines. Basically, this involved the design and development of a reliable, inexpensive, and effective outboard-engine-power-takeoff system to drive a hydraulic pump.

The initial application of this hydraulic system was directed to the salmon troll fishery on the Oregon Coast, with the Pacific City-type dory selected as the fishing test platform. Figure 1 shows two typical Pacific City dories. These vessels cost from $800 to $1200 and are rugged sea boats, but they have been limited in productivity because the trolling lines were fished by hand-powered gurdies or reels. Figure 2 illustrates a typical dory equipped with hand-powered salmon gurdies. Despite this limitation, these dories offer a potential in the fishery that has not yet been fully exploited. When powered by 50- to 70-hp engines, the 20- to 22-foot dories attain speeds of 18 to 30 mph, providing high-speed-at-sea capability for broader fishing-ground coverage. In addition, they can be transported by trailer overnight to different fishing grounds along the coast. A secondary objective was the development of a specialized hydraulic system for use with outboard-engine-driven boats, particularly, the dory type craft described above.
Figure 1. Two typical Pacific City type dories.

Figure 2. Typical dory equipped with hand-powered salmon gurdies.
Scope of Project

The project intention was to ultimately develop a typical system configuration which could be adapted to any outboard engine in the power range of 40 to 70 hp. Three hydraulic systems: systems A, B, and C, were designed, developed, and tested throughout the 1970 summer salmon fishing system on the Oregon Coast. System D, a fourth and considered optimum system, was designed but not actually tested under operational fishing conditions as the development was subsequent to the summer fishing season.

Four engines were made available to the project by their respective manufacturers. System A, the prototype system, was built around a 1970 model 60 hp Johnson engine furnished by Outboard Marine Corporation.

System B was powered by a 1970 model 55 hp Bearcat engine furnished by the Fisher-Pierce Company. Two versions of system C were developed; the first powered by a 1970 model 50 hp Mercury engine furnished by the Mercury-Kiekhafer Corporation, and the second by a 1970 model 40 hp Johnson engine also furnished by Outboard Marine Corporation.

Each outboard motor presented unique problems in designing a pump mount. The pump mount in each instance had to be securely anchored onto the engine blocks, and the top of the outboard shroud
altered to allow space for the installation. Hydraulic components had to be selected carefully to obtain equivalent performance from four different engines.

It soon became apparent that the salmon gurdies used on larger boats were not appropriate for dories or other small high-speed outboard-powered trollers, as the commercially available gurdies were too expensive (approximately $1,250 for four spools complete) and too heavy.

Accordingly, the project also sought to develop less expensive and lighter gurdies appropriate for dories. Experience acquired as each dory was rigged was incorporated into each succeeding rig. This evolutionary process resulted in a compact, light, and relatively inexpensive gurdy system for outboard-powered trollers.
II. BASIC SYSTEM DESIGN

Operational Requirements

The three parameters to be determined in designing a hydraulic-powered-fishing-machinery system to be powered by the boat's outboard-engine drive unit are: 1) the work cycle the fishing operation requires, in this case, gurdy line speed requirements and maximum line tension necessary to operate fishing gear properly; 2) the horsepower required from the main propulsion unit to power the gear; and 3) the surplus power available from the outboard engine to drive a hydraulic pump. It was deemed undesirable to reduce engine speed by more than approximately 50 rpm by the removal of surplus power.

Consideration of the first parameter, the establishment of appropriate work cycles for various types of fishing machinery resulted in the following design requirements. Salmon trolling gurdies: average of 250 feet per minute line speed; maximum total of 100 pounds line tension. Tuna pullers: average of 360 feet per minute line speed; maximum of 125 pounds line tension. Longline haulers: average of 120 feet per minute line speed; maximum of 250 pounds line tension. Pot, trap, or ring haulers: average of 90 feet per minute line speed; maximum of 300 pounds line tension.

These line speed and pulling force requirements were those
judged appropriate for small outboard-engine-powered boats such as dories, and are not necessarily appropriate for larger fishing boats with slower inboard engines.

The second consideration was to determine the power required from the engine to turn the hydraulic pump. A system incorporating salmon trolling gurdies was selected as the prototype system for design purposes.

The first step in this determination was consideration of the work cycle previously specified for a salmon trolling system; a gurdy line speed of 250 feet per minute and a line tension of 100 pounds force. It was known that the engine trolling speed of a 22 foot long dory would average 1800 rpm when driven by a 60 hp outboard engine.

The variables for each spool were:

- Torque (inch-lbs.)
- Line tension (lbs.)
- Effective spool radius (inches) (3 inches for salmon gurdies)
- Spool rotational speed (revolutions per minute)
- Constant (3.14)
- Line haul speed (inches per minute)

\[
T = PR = 100 \times 3 = 300 \text{ in-lb.} \quad (2-1)
\]
Once the required torque (300 inch pounds in this instance) and the spool rotation speed in rpm (approximately 160 rpm in this system) were known, the power requirement was determined for the hydraulic system by using standard formula (2-3).

Under ordinary fishing circumstances no more than two spools would be operating simultaneously. The system would require two spools concurrently at 300 inch pounds of torque per spool.

\[
S = \frac{L}{2 \pi R} = \frac{3000}{2 \times 3 \times 3.14} \approx 160 \text{ RPM} \quad (2-2)
\]

Thus for two spools:

\[
HP = 2 \times \frac{300 \times 160}{63,000} = 2 \times 0.762
\]

\[
HP = 1.52
\]

All hydraulic systems lose some power because of the friction generated by the oil passing through the system. Typically, a hydraulic system is approximately 83% efficient (\(\eta = 0.83\)). The power requirements to operate the pump can be expressed as:

\[
\frac{HP}{P} = \frac{HP}{\eta} = \frac{1.52}{0.83} = 1.83 \quad (2-4)
\]

These calculations established that a maximum of 1.83 horsepower was required from the outboard engine operating at about
1800 rpm for the prototype system.

The third parameter, the available surplus engine power, was established by studying the engine brake horsepower and propulsion horsepower versus rpm curves of a typical outboard engine used to power dories.

The full-throttle brake horsepower versus engine rpm curve was supplied by the Outboard Marine Corporation. This curve is presented in Figure 3. In addition, a propulsion horsepower versus rpm curve is also illustrated in Figure 3. This curve was developed by making the approximation that the power required to propel a boat is proportional to the engine rpm to the third power (5, p. 382-387). This approximation neglects certain propeller losses and other losses present at very high rpm but gives reasonable values, especially in the rpm range of interest, 1200 to 2500 rpm.

The difference, $\Delta P$, between the brake hp and the propulsion hp curve at any point represents the surplus power available to power auxiliary equipment, in this case a hydraulic pump. It must be noted that the engine brake horsepower curve is a full-throttle curve and the engine would not ordinarily be operated at full throttle while fishing gear is being used, although it is apparent that at any throttle setting, more than 2 hp is available over that required to propel the boat.

When additional power is required from an engine operating at
Figure 3. Power versus rpm--60 hp Johnson outboard engine.
a particular throttle setting, the engine responds by slowing down
and establishing a new operating rpm. In the case of an outboard
engine, the power required to propel the boat is less at the new rpm.
The engine then is developing power equal to the sum of the additional
power required to operate the pump, in this case, and the power re-
quired to propel the boat.

For design purposes it was assumed that the use of an additional
2 hp from an engine would not significantly reduce rpm. Subsequent
operational tests did indicate that this assumption was reasonable.
These tests revealed that engine speed reductions were not in excess
of 50 rpm for all engines used.

**System A Configuration**

Consideration of the previously mentioned requirements dictated
that the system as conceived should be capable of delivering a max-
imum of 1.52 horsepower to the salmon gurdies. The gurdies would
be required to turn at a maximum of 150 rpm.

The overall configuration was quite standard and similar to
that used on larger fishing boats powered by automotive type engines.
The primary difference between the prototype system and existing
systems was the manner in which the hydraulic pump was installed
on the engine. Minor differences involved lighter weight and less
complex circuitry.
Figure 4 illustrates the general configuration of system A as originally conceived and constructed. The major components are designated symbols A through E. "A" indicates the salmon trolling gurdies, one set on each side of the boat, positioned approximately amidships. Each gurdy set consists of two spools and associated hardware. System A utilized gurdy sets powered by a single hydraulic motor in each case. Subsequently developed gurdy sets, were powered by one motor per spool. "B" indicates the hydraulic pump mounted on the outboard motor; "C" indicates the oil storage reservoir; "D" indicates the control valves, one for each gurdy set; and "E" indicates the oil filter. The arrows represent the oil lines and indicate flow direction.

Figure 5 illustrates a detailed schematic diagram of system A. Standard hydraulic symbols indicate the components described above.

Material Selection

Utilizing materials resistant to corrosion in a salt water environment was considered important. The corrosive potential of sea water is well known and understood, and most non-alloyed, ferrous materials are subject to corrosion in this environment. The obvious solution to the corrosion problem was to make use of corrosion resistant alloys and possibly non-metallic materials, but the achievement of a relatively low cost installation was important.
Figure 4. System A--general configuration.
Figure 5. System A--schematic diagram.
Three methods are commonly used to reduce corrosion in any hostile environment: use of corrosion resistant alloys; use of coatings; and the application of cathodic protection. For this project, a combination of the first two methods was used. Due to the electrical isolation of most of the hydraulic system components, the use of cathodic protection was deemed impractical.

Initially a clad aluminum alloy (AISI 7050) was used for fabrication of mounting brackets and other low stress components. All metallic components were coated with a multilayered application of zinc chromate paint as a primer and at least two layers of an epoxy based marine paint.

Several plastics were used in the project especially on the systems developed subsequent to the prototype system. Plastics resistant to the undesirable effects of solar radiation are required in this type of application.

Problems developed during testing of a subsequently developed system (system B) due to the flexibility of the aluminum alloy used to fabricate the hydraulic pump mounting bracket. This mount was subject to excessive vibration especially while the engine was operating in the cruise mode (4000 to 5000 rpm) and a fatigue failure did occur in the pump mounting bracket.

A new mounting bracket was fabricated from cold-rolled steel plate and when well protected with paint was considered quite
satisfactory. Subsequently developed brackets for the other engines used were also fabricated from steel plate and were found to be satisfactory.

**Hydraulic Pump Installation**

With the basic system requirements for an outboard-engine-powered hydraulic system established, the next step involved the selection of an appropriate pump and the design of the pump installation.

Two basic configurations were considered. The first consisted of a mount design where the pump would be driven by a belt and pulley arrangement from the outboard flywheel. The second configuration considered was a direct-drive installation with the pump mounted with its shaft axially in line with the outboard engine crankshaft.

Each of the configurations presented unique problems. Both approaches shared the common problem of inadequate space under the outboard motor shroud for a pump installation. The wide range of engine rpm of the outboard motors necessitated using a pump with high rpm capability. The top of the crankshafts in all outboard engines are not supported by bearings strong enough to bear the side loads which would be generated by belt forces. Finally, the mount system and pumps had to be installed in a manner that would not impart or receive excessive vibrations under
operating conditions.

Throughout the design phase the original requirements of relatively low cost, weight and space conservation, simplicity, and effectiveness further conditioned design configurations. These requirements led to a rejection of clutch-activated drive systems.

Information supplied by the outboard companies indicated that additional outboard bearing supports would be necessary to protect the crankshaft upper main bearing if a belt-driven power-take-off was employed. If these outboard bearings were not built into the power take-off, it was felt that bearing and oil seal failure might well result.

Consideration of all the above factors resulted in the selection of the less complex and more compact direct-drive configuration, with the pump being mounted axially in line with the crankshaft. The pump shaft and the shaft extension bolted to the flywheel were to be joined by a coupling. Figure 6 illustrates the general layout of this prototype installation.

A pump had to be selected that was of small size, relatively light weight, and which could be operated under no-load conditions to 5500 rpm.

After considerable study, a P-1 Series pump manufactured by Tyrone Hydraulics, Inc., was selected. This pump is available in four models with outputs up to 13.3 gallons per minute (gpm), speeds to 6000 rpm, and is capable of operating pressures of up to 3000
Figure 6. Prototype outboard engine hydraulic pump installation.
pounds per square inch (psi).

The Tyrone Model P-1-25 Jobmaster pump provides a hydraulic fluid displacement of approximately 3.6 gpm at 1600 rpm, 4.0 gpm at 1800 rpm, 4.4 gpm at 2000 rpm, and 5.0 gpm at 2200 rpm. These gpm displacement figures are more than adequate to power the hydraulic motors in the fishing machinery systems discussed below. Figure 7 illustrates the displacement versus rpm characteristics of this pump.

A Morse N Series Delron flexible coupling was chosen to connect the pump shaft and engine flywheel shaft extension. This coupling is capable of withstanding speeds of 5000 rpm under load and will transmit 6.5 hp at 1800 rpm. Since the required horsepower to drive the pump at 1800 is only 1.83 hp maximum, an adequate safety margin is assured. The Morse coupling has two further advantages. The Delron chain can be quickly removed from the sprockets and links can be changed by driving out the appropriate pins and inserting new links. The flex coupling will also tolerate an angular misalignment of up to 1/2 minute between shafts.

The design of the hydraulic pump mounting for the 60-hp Johnson outboard engine was rather straightforward, as mounting points are readily available on the rear of the engine block and a reasonable amount of space is present under the motor shroud. This mount was fabricated from 1/4-inch Alclad 7050 aluminum alloy,
Figure 7. Hydraulic pump speed versus displacement—Tyrone model P-1-25.
although, as was discussed previously, this material was not found to be suitable for this application. Figure 8 details the final design developed for the mounting components of system A. Figures 9 and 10 present photographs of the pump installation on the 60-hp Johnson engine. Note the Kolstrand salmon gurdies in the upper left background in Figure 9. During the fabrication and installation of the shaft extension plate, care was taken to ensure that the flywheel top surface was perfectly true to the crankshaft axis. To meet this requirement, the flywheel was turned in a lathe and its top surface machined slightly. The underside of the extension shaft plate was machined to form a recess for the flywheel locking nut which protrudes slightly about the flywheel. A slight mating and centering surface was cut on both the flywheel upper surface and extension plate surface. Mating and centering surfaces thus ensured that the pump extension plate shaft and the engine crankshaft were axially aligned. The extension plate was bolted to the outboard flywheel using the existing tapped flywheel puller holes on the flywheel and the appropriate-sized machine bolts.

After the shaft extension was installed on the flywheel, the whole assembly was rotated in a lathe and checked for trueness.

It was found during fabrication and later testing that certain critical procedures must be followed for a satisfactory installation. The prime requirement is proper axial alignment of the pump shaft
MOUNTING BRACKET

NOTE:
1. BRACKET ¼ MILD STEEL
2. ALL DIMENSIONS ± ¼ (0.030"
3. FINISH 2 COATS ZINK CHROMATE PRIMER
2 COATS MARINE EPOXY PAINT
4. ALL COMPONENTS TO BE CUSTOM FITTED TO ENGINE
5. WELD GUSSETS IN PLACE AND GRIND FLUSH

Figure 8. Hydraulic pump mounting components---60 hp Johnson.
Figure 9. Prototype pump installation--rear view.

Figure 10. Prototype pump installation--front view.
and the engine crankshaft for vibration-free operation and long component life. The angular misalignment of these shafts must not exceed 1/2 minute.

The pump mount bracket was cut to fit the contours of the rear of the engine block and then bolted onto solid surfaces on the block itself. This resulted in a rigid mount relatively free of vibrations. Four bolt holes were drilled in the rear leg after the axial alignment of the pump shaft and extension shaft had been checked. The prototype mount was bolted onto the engine head by using the two uppermost head bolts and two existing attachment points for the original engine lifting ring. Appropriate spacers were used to secure a rigid mount.

The Morse Delron flexible coupling sprockets were keyed onto each shaft. The flex coupling sprockets were held in place by set screws tightened onto the keys. An additional set screw was placed in each sprocket at a 90° angle from the first for extra security. The sprockets were separated by 1/4 inch to guard against binding and to allow the Delron coupling to absorb minute misalignments.

The top was cut out of the outboard motor cover to accommodate the pump and bracket protruding above the flywheel. A protective cover was fabricated from 16-gauge aluminum sheet to cover this opening on the 60-hp motor. A photograph of this cover is illustrated in Figure 11. This cover was then fastened to the shroud with sheet metal screws.
Finally, two 1 1/2-inch holes were cut in the lower shroud of the engine for the suction line from the reservoir to the pump and the high pressure line from the pump to the gurdies. Rubber grommets placed in the shroud holes to protect the hose from chafing are shown in Figure 11.

![Prototype pump protective cover installed.](image)

**Figure 11.** Prototype pump protective cover installed.

**System Components**

In addition to the previously described pump assembly, the other parts of the system were carefully selected bearing in mind the
need for light weight, durability, and relatively inexpensive equipment. Referring to Figure 4, the following components were selected: A, 2-spool Kolstrand gurdies which includes one hydraulic motor and two clutches per gurdy; B, hydraulic pump; C, Char-Lynn R244 three gallon reservoir; D, Gresen SPW 4 control valves with built-in relief valve; E, Gresen FA 103 filter; and Aeroquip medium pressure hose.

Subsequently developed systems used the same model reservoirs, pumps and oil filters but other types of valves and gurdies. All gurdies used were powered by Char-Lynn model AC hydraulic motors. These motors were selected after considering the torque and rpm requirements of the salmon gurdies.

Thermal Considerations

The inefficiency in a hydraulic system is due to pressure reductions where no useful mechanical work is done. Primarily, these pressure reductions are due to oil flowing through relief valves, and friction in the system. The energy lost by the oil due to these pressure reductions appears in the system as heat.

Testing of system A under actual fishing conditions revealed the following information on oil pressures: At normal engine trolling speeds, 1500 to 2000 rpm, the pump discharge pressure averaged 200 psi when a gurdy was pulling in a single fishing line equipped with standard fishing gear. No more than two lines were ever pulled in
simultaneously, consequently maximum pump discharge pressure
did not exceed 600 psi even with many fish on each line. Pump dis-
charge pressure under no-load conditions was 50 psi with the engine
operating at 5500 rpm and 5 psi at 1800 rpm.

Essentially, the heat generation during system operation under
maximum load conditions is equal to one minus system efficiency
multiplied by the power input to the pump. During no-load operation,
the energy in the pump discharge pressure is almost completely
converted to heat according to the relationship (1, p. 150) \( Q_G = (1.5)(\Delta \text{psi})(\text{gpm}) \).

For system A:

1. **Underload**

   \[
   Q_G = (1.0 - \eta) \frac{\text{HP}}{\text{p}} \\
   Q_G = (1.0 - 0.83) \times 2.0 \\
   = 0.34 \text{ Hp} \\
   = 0.34 \text{ Hp} \times 2545 \frac{\text{BTU}}{\text{HP} \cdot \text{Hr}} \\
   \eta \text{ System efficiency (0.83)} \\
   Q_G = 866 \text{ BTU/Hr} \\
   \text{HP} \text{ Horsepower input to pump}
   \]

2. **No-load conditions**

   \[
   Q_G = (1.5)(\Delta \text{psi})(\text{gpm}) \\
   \]

   a. 1800 RPM

   \[
   Q_G = 1.5 \times 50 \times 13 \quad \Delta \text{psi system pressure drop} \\
   = 975 \text{ BTU/Hr}
   \]
The heat dissipation from this type of system is almost completely dependent upon convective transfer from the storage reservoir. A small amount of heat is also lost by the other parts of the system. The governing equation for this heat transfer mechanism is:

\[ Q_c = \bar{h}_c \Delta T \]  

(2-7)

- \( Q_c \) = Rate of heat transfer by convection in BTU/Hr
- \( \Delta T \) = Difference between surface temperature and the ambient temperature of the surrounding air. \( T_s - T_a \)
- \( \bar{h}_c \) = Average convective heat transfer coefficient
  - BTU/Hr - ft\(^2\) - °F

**Five-gallon reservoir**

Effective reservoir surface area = 4.5 ft\(^2\)

Effective surface area of hose, valving, motors, etc. = 0.5 ft\(^2\)

Total Area = 5.0 ft\(^2\)

\( \Delta T = T_{\text{surf}} - T_{\text{air}} \)

Average surface temperature \( T_s = 80 \) °F

Average summer Oregon Marine air temperature \( T_a = 50 \) °F

\( \Delta T = 30 \) °F

For air free convection (6, p. 15) \( h_c = 5.0 \ \frac{\text{BTU}}{\text{Hr} \cdot \text{ft}^2 \cdot \circ F} \)

\[ Q_c = 5.0 \times 5.0 \times 5.0 \times 30 \]

\[ Q_c = 750 \ \text{BTU/Hr} \]
The average heat generation for this system is approximately the time averaged sum of the heat generation in the different operational modes.

Consider an operational day of 12 hours:

<table>
<thead>
<tr>
<th>Mode--Time</th>
<th>Heat Generation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cruise no-load 3 hrs.</td>
<td>$975 \times 3 = 2925 \text{ BTU}$</td>
</tr>
<tr>
<td>Trolling no-load 6 hrs.</td>
<td>$30 \times 6 = 180 \text{ BTU}$</td>
</tr>
<tr>
<td>Trolling underload 3 hrs.</td>
<td>$866 \times 3 = 2598 \text{ BTU}$</td>
</tr>
</tbody>
</table>

Average heat generation = 475 BTU/Hr

Therefore, since the average heat generation is less than the average heat dissipation by convection, i.e., 465 BTU/Hr vs. 750 BTU/Hr, no temperature build-up should occur over long time periods. The higher heat generation during system operation under load and at cruise would cause a short-term rise in oil temperature above the steady state value, but this rise would be slow and could be ignored.

The reservoir must be mounted in such a way as to allow free air flow to all reservoir surfaces for heat dissipation purposes. The heat dissipation must be high enough to keep the oil temperature below 150 ° at all times. Oil temperatures above this point can cause undesirable changes in the oil viscosity and also deterioration of system components. Adherence to the principle of mounting the
reservoir on strips of firring and then fastening the firring to the hull or transom allows ample air circulation and ensures heat dissipation capability.

Operation in colder ambient temperatures would, of course, significantly increase heat dissipation, but operation in higher ambient temperatures (in the tropics for example) would possibly require the installation of an additional heat exchanger.

Operational testing of the prototype and following systems confirmed the above analysis to be quite reasonable. At no time did oil temperatures exceed 150°F. Hydraulic oil temperatures averaged approximately 100°F on all four systems tested, fluctuating around this point by about 20°F depending upon operating conditions.
III. SYSTEM DEVELOPMENT

System B

The second system developed was patterned basically around System A. The major difference was the type engine used, a 55 hp Fisher-Pierce Bearcat. Several minor changes were incorporated, each the result of testing, and suggestions by the operators of the prototype system.

An adjustable flow control valve was included in the system to provide constant flow and consequently constant gurdy speed regardless of engine speed above about 1000 rpm. The maximum oil flow could be changed to suit the individual fisherman. This addition made the system much more convenient to operate during fishing operations.

The pressure relief valve used in this system is independent of the gurdy control valves.

A slightly different type of gurdy was used in system B. This type gurdy shown in Figure 12 is manufactured by Jim Lyons of San Diego, Calif. and was originally intended for tuna fishing but adapted well to the salmon trolling methods. It was included primarily due to its reduced cost and weight. Figure 13 illustrates the approximate layout of system B including components (D) and (F), the flow control valve and the relief valve respectively. The position
of these two additional components is also shown in Figure 14, the
detailed schematic of the system. The details of the hydraulic pump
mounting bracket, and the shaft extension for the Bearcat engine are
reproduced in Figure 15. This design was very satisfactory due to
its rigidity and ease of fabrication.

Figure 12. Lyons tuna-salmon gurdy.
Figure 13. Hydraulic system B--general configuration.
FLOW CONTROL VALVE, PRESSURE COMPENSATED

CONTROL VALVE 1.

MOTOR 1. OPERATES 2 PORT GURDY SPOOLS

CONTROL VALVE 2.

MOTOR 2 OPERATES 2 STARBOARD GURDY SPOOLS

ALL CONTROL VALVES FULL REVERSIBLE WITH SPEED CONTROL

Figure 14. Hydraulic system B--schematic diagram.
NOTE,

1. THIS PIECE MACHINED FROM ONE SOLID BLOCK OF STRESS-PROOF STEEL
2. ALL DIMENSIONS ± 0.030
3. ALL COMPONENTS TO BE CUSTOM FITTED TO ENGINE
4. FINISH - 2 COATS ZINK CHROMATE PRIMER
   2 COATS MARINE EPOXY PAINT

NOTES:
1. FABRICATION FROM 1/2" MILD STEEL
2. ALL DIMENSIONS ± 0.030
3. ALL COMPONENTS TO BE CUSTOM FITTED TO ENGINE
4. FINISH - 2 COATS ZINK CHROMATE PRIMER
   2 COATS MARINE EPOXY PAINT

Figure 15. Hydraulic pump mounting components -- 55 hp Bearcat.
System C

This system is quite similar to system B. The main difference involved the use of gurdies especially developed for this project instead of the commercially available sets previously mentioned. Each of these gurdy sets consisted of two hydraulic motors, each motor connected to an injection-moulded polyethylene spool. Both of the spool units shared a common mounting bracket. Control valves for each motor were also mounted on the bracket. A view of these gurdies is presented in the left and right sides of the photo in Figure 22. The system C gurdies have no clutch mechanism and consequently are controlled completely by valves, one per motor. Figure 16 shows detailed schematic of the system. Note the four control valves, one per hydraulic motor.

System C was designed to be compatible with two different outboard engine power plants. Two installations were made, one designed around each engine.

1. Fifty hp Mercury outboard engine

Figure 17 details the pump mounting components for this engine. The configuration of this pump mounting bracket turned out to be very satisfactory due to its triangular shape and consequential rigidness.

Figure 18 shows a view looking down on the installed pump. Note the tripod style mounting bracket. Figures 19 and 20 show the
OUTBOARD MOTOR

FLOW CONTROL VALVE, PRESSURE COMPENSATED

CONTROL VALVES 1-4

REVERSIBLE MOTORS #1 WITH SPOOL

#2 WITH SPOOL

#3 WITH SPOOL

#4 WITH SPOOL

FILTER

RESERVOIR

PUMP

ALL CONTROL VALVES FULL REVERSIBLE WITH SPEED CONTROL

Figure 16. Hydraulic system C—schematic diagram.
NOTES:
1. MOUNTING BRACKET - MILD STEEL
2. ALL DIMENSIONS ± 0.030
3. USE APPROPRIATE SPACERS BETWEEN BRACKET AND MOTOR
4. FINISH - 2 COATS ZINC CHROMATE PRIMER
   2 COATS MARINE EPOXY PAINT
5. SHAFT EXTENSION MACHINED FROM ONE SOLID PIECE OF STRESS-PROOF STEEL

Figure 17. Hydraulic pump mounting components -- 50 hp Mercury.
pump installation on the 50 hp Mercury engine with the pump cover off and in place. In Figure 20, note the pump protective cover and the oil hoses passing through one side of it. In the right side of Figures 20 and 24 the oil reservoir, flow control valve, and pressure relief valve are visible. Note the quick-release hose couplings in Figure 20. These couplings were used in system A and on the Mercury engine in system C. They were deleted in the other systems to reduce cost.

Figure 18. Fifty hp Mercury outboard engine with hydraulic pump installation.
Figure 19. Fifty hp Mercury pump installation—protective cover off.

Figure 20. Fifty hp Mercury pump installation—protective cover in place.
Figures 21 and 22 are presented to give overall views of system C powered by the 50 hp Mercury engine installed on a typical Pacific City dory. Note the gurdy sets on each side of the boat in Figure 21.

Figure 21. System C—50 hp Mercury—viewed toward stern.
2. Forty hp Johnson outboard engine

The pump installation on this engine was similar to the 60 hp Johnson configuration previously discussed. Problems developed with this installation during testing. The 40 hp Johnson engine is shock mounted in a rubber bushing arrangement. Engine vibrations are not damped by the boat transom so the engine moves about on its mount. This shaking motion caused high inertial loads upon the original pump mount installed and caused several minor structural failures. Consequently, the mount was modified to make it as rigid as possible. This arrangement as shown in Figure 23 was found to be satisfactory. In Figure 24, note the system C gurdy set just to
the left of the outboard engine and the pump protective cover installed on the engine. Figure 25 illustrates a more satisfactory design for the 40 hp Johnson pump mounting bracket. This design was not fabricated and tested, but based on its similarity to the successful 60 hp Johnson mounting bracket design, was deemed optimum for this engine.

Figure 23. Forty hp Johnson modified hydraulic pump installation.
Figure 24. Forty hp Johnson pump installation--protective cover in place.
Figure 25. Hydraulic pump mounting components—40 hp Johnson.
System D

The last, and considered optimum, hydraulic system designed during this project is depicted schematically in Figure 26. Note the integral flow control and pressure relief valve incorporated in the system. The primary difference between this system and system C involves the improved gurdy sets and the slightly different valving setup. In cooperation with the Lauterbach Company of Portland, Oregon, these light weight gurdy sets were developed as a project objective. These gurdies have polyethylene spools and chassis. The motors are standard steel units and standard control valves are used, but the overall weight for two sets of gurdies is about 145 pounds installed. This is almost a 50% reduction in weight over commercially available gurdy sets. Figure 27 illustrates this gurdy set with plastic chassis, spools, and convenient control valve location.

The pressure relief valve and flow control valve are incorporated in this system, thus reducing system cost and complexity. These gurdy sets were not actually tested but are being commercially produced and marketed by the Lauterbach Company at this time for the upcoming fishing season. They are considered an improvement over the system C gurdy sets due to the decreased weight, cost, and susceptibility to corrosion. The basic design and components were tested thoroughly in system C.
Figure 26. Hydraulic system D—schematic diagram.
Figure 27. Lauterbach gurdy set.
Other Fishing Uses

The use of these systems is directly applicable to such uses as: Long line fishing for bottomfish, fishing for albacore tuna, trap and pot fishing, and gillnet fishing.

There is a significant latent demand for hydraulic powered equipment driven by outboard engines in these areas. The same general procedures and methodology used in the design of the salmon trolling system can be used to rate the work cycles of systems for other fishing gear.

A continuing effort is going into further development directed toward these other areas. Fishing equipment specialists at the Oregon State University Marine Science Center and private interests are deeply involved in this effort at this time.
IV. PROJECT RESULTS AND CONCLUSIONS

Evaluation

The four dories working on the hydraulic project logged a total of approximately 2,724 hours during the 1970 salmon season. Two of the engines were dismantled at the conclusion of the project and no signs of unusual wear were noted on any of the crankshafts, main bearings, connecting rods and bearings or other engine parts. The relatively trouble-free operation of all of the engines once initial application problems were overcome supports an over-all conclusion that the hydraulic power take-off system is safe, feasible, and efficient providing the following principles are observed:

1. Care must be taken during the installation of the pump to ensure that the shaft extension is axially aligned with the crankshaft. No wobble of the shaft extension can be tolerated.

2. The pump mounts employed must be rigid and the pump mount attachments must be of such a nature that bolts fastening the pump mount to the engine block will not work loose under engine operation.

3. These bolts should be checked from time to time to ensure that they remain securely fastened. It was found during the life of the project that bolts properly fastened with lock washers did not work loose.
4. Good hydraulic practices should be followed, such as checking each major component in the system by an operational test before installation to assure that each component is working according to specifications. Good installation practices should be observed, such as the inclusion of a filter in the system as dirty oil can shorten the life of pumps, control valves, and motors. All components, especially hoses and fittings, should be inspected for cleanliness prior to system installation and cleaned if necessary. A high-quality hydraulic fluid should be used in the system.

Project Economics

A breakdown of the project testing time reveals the economic potential of this type system. A total of approximately 2725 hours of operating time was put on the systems. Table 1 breaks this down per system in terms of operating hours, gross earnings, and earnings per hour for the system dories and for two randomly selected, hand-powered-gurdy-equipped dories.
Table 1. Dory project economic data.

<table>
<thead>
<tr>
<th>System</th>
<th>Powered Systems</th>
<th>Hand Powered Systems</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Prototype A B</td>
<td>C-1 C-2</td>
</tr>
<tr>
<td>Operating hours</td>
<td>1275 550</td>
<td>350 550</td>
</tr>
<tr>
<td>Earnings ($)</td>
<td>15,000 4,700</td>
<td>3,200 4,300</td>
</tr>
<tr>
<td>Earnings per hour ($)</td>
<td>8.50 8.60</td>
<td>9.10 7.80</td>
</tr>
</tbody>
</table>

The figures in the above table reveal that the hydraulic-powered-fishing equipment earned approximately $8.50 per hour per dory. The two dories equipped with hand-operated gurdies picked from the Oregon Dory Fleet at random earned 6.8 and 4.2 dollars per hour each. This yields an average earning of approximately 5.5 dollars per hour for dories equipped with hand-powered salmon gurdies.

It must be mentioned that the experience and skill of the individual fisherman plays an important role in the yield that is produced by the fishing operation. Powered fishing gear increases the efficiency of a small commercial boat a great deal but will not significantly increase productivity unless the operator has a reasonable amount of skill, experience, and drive to make maximum use of the gear's potential. These variables are apparent upon examination of the above production figures.

The scatter in the earnings per hour figures is due to the fact that although systems B and C were operated less than 50% of the
time system A was operated, a high percentage of this time was put on systems B and C at the peak of the salmon run. System A operated throughout the season and consequently had many days of low production due to the absence of fish schools.

The material cost of each system used during this project was in excess of $1000 but the final configuration as developed, system D, will bring the cost per system to somewhat less than $1000. A commercial hydraulics firm, the Lauterbach Co. of Portland, Oregon is presently marketing the system developed during the project as a kit for about $875.00. This does not include the installation cost of the system but includes all components needed to set up a four-spool salmon fishing system.

**Future Uses**

It has been effectively demonstrated on a limited scale that this type of system is definitely feasible from both an economic and engineering viewpoint. Future adaptations of the concept will establish its large-scale economic feasibility.

The advantages of these systems suggest strongly that they will have world-wide application in the near future as the demand for increasingly efficient food harvesting methods develops due to the exponentially increasing world population.

Much interest has been shown in this project by various
agencies and literally hundreds of individuals on a world-wide scale. The U. S. State Department, U. S. Bureau of Commercial Fisheries, and several foreign governmental agencies have shown a strong interest in this type of equipment for use in the emerging nations of the world.

It has the distinct advantages of inherent simplicity and low cost while enabling a small boat to compete in a limited fashion with much larger and more expensive craft.


