STUDY OF STORED ENERGY SYSTEMS PROPOSED FOR TESTING A PRESSURE-REGULATING VALVE(U) ARMY ENGINEER WATERWAYS EXPERIMENT STATION VICKSBURG MS STRCC J B CHEEK UNCLASSIFIED JUL 86 WES/MP/SL-86-6
STUDY OF STORED ENERGY SYSTEMS PROPOSED FOR TESTING A PRESSURE-REGULATING VALVE

by

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July 1986
Final Report

Approved for Public Release, Distribution Unlimited

Defense Nuclear Agency
Washington, DC 20305

Work Unit 31588
Study of Stored Energy Systems Proposed for Testing a Pressure-Regulating Valve

Cheek, James B.

Final report from July 1986 to July 1986

Two systems using stored mechanical energy, rotational and translational (drop), were proposed for use in testing a high discharge, high-pressure-regulating valve. A computer-based dynamic analysis of those systems indicates the drop test system to be less costly, but near the practical limits of drop height and weight. The rotational system will exercise the valve for one-half of a pump cycle provided the pump begins at the start of a pump output cycle. The rotational system must be able to withstand very large forces. Neither system exercises the valve for enough time, but the rotational system appears to be capable of longer testing time.
Non-SI units of measurement used in this report can be converted to SI (metric) units as follows:

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<td>gallons (US liquid)</td>
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SUMMARY

Two valve test systems were studied. They each use a stored energy approach; one is rotational, the other is translational (drop). An idealized dynamic analysis indicates the rotational system will generate peak forces and torques that will require a massive test fixture. The test fixture needs a thorough design and dynamic analysis to assure it will operate under such severe dynamic forces.

The drop test will almost equal the performance of the rotational system. It also appears to be less costly. However, it is near the practical limits of ball weight, drop height and available reaction structure.

Neither system produces the design discharge for enough time to evaluate the regulating valve's performance. Precise timing of the start of pumping in the rotational test is critical to attaining the desired discharges.
This study was conducted in January 1986, by personnel of the US Army Engineer Waterways Experiment Station (WES), under the sponsorship of the Defense Nuclear Agency (DNA) in support of the Silo Test Program-Shock Isolation Systems. The DNA project officer was Mr. James Cooper. Mr. Larry Selzer, Aerospace Corporation, proposed the concept as a means of evaluating a valve for a full-scale shock isolation system.

The investigation was conducted under the supervision of Messrs. Bryant Mather, Chief, Structures Laboratory (SL); James T. Ballard, Assistant Chief, SL; Dr. Jimmy P. Balsara, Chief, Structural Mechanics Division (SMD), SL; and Mr. Robert E. Walker, Project Manager, SMD. This report was prepared by Mr. James B. Cheek, SMD.

The Director of WES during the investigation and preparation of this report was COL Allen F. Grum. The Technical Director was Dr. Robert W. Whalin.
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<tr>
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PART I: INTRODUCTION

1. The analysis documented in this report was accomplished as part of a feasibility study to develop design requirements for a device proposed for testing a hydraulic pressure regulating valve. Because of the valve's high operating pressure and discharge, designing the test device presents many difficult analysis problems. This study looks at but one of those problems in a highly idealized fashion. Nevertheless, the analysis is useful in that it establishes the best performance attainable from a "perfect" system. That performance can be used to see how well it meets test system requirements. From that evaluation, decisions on any changes needed to a practical system can be made and more extensive engineering analysis can be conducted.

PART II: IDEALIZED ANALYSIS, ROTATIONAL

Valve Specification

2. The following analysis was done on a fixture proposed to test a regulating valve at a constant regulating pressure of 4,350 pounds per square inch (psi), at a design maximum flow of 26.4 gallons per second (gps).

Test Fixture Data

3. The Test Fixture consists of two, 30 inch diameter, solid disc flywheels weighting 1,000 pounds (lb). Each is connected to a crankshaft having a one-inch offset (throw) which is in turn linked by a connecting rod to a pump as shown in Figure 1 below. The pump cylinder's ID is 16 inches and the pump shaft's OD is 9 inches.

![Diagram of Test Fixture](image-url)
Dynamic Analysis

4. The analysis is based on a perfect system (no mechanical or pumping loss). The results, presented in Appendix A, indicate the following:
   a. At 424 revolutions per minute (rpm), the pump will discharge 26.4 gps, peak, provided the pumping starts when the crankshaft angle is 90°. The discharge will decline, as shown in Figure A2, to zero in 71 milliseconds (msec).
   b. Peak torque in the drive is near 50,000 foot-pounds (ft-lbs) (see Figures A1 and A3).
   c. The drive will slow from 424 rpm to zero rpm in 70.8°/360° = .197 revolutions.
   d. The force required to operate the pump shaft is almost 600,000 lb at design pressure.
   e. Operating the system at 577 rpm and starting the test at zero degrees crankshaft angle produces the desired peak discharge and increases the time of regulated discharge (See Figure B7).

5. Appendix B, like Appendix A, graphs the test fixture performance. However, pumping starts at a crank angle of zero degrees. Peak pump output is only 16 gps using 424 rpm as above (see Figures B1 and B2).

6. A second series of calculations was made keeping all conditions the same except the shaft speed which was changed to 577 rpm in order to raise the peak pump discharge to near the design specification (26.4 gps). The results of those calculations are presented on Figures B7 through B12.

7. Those calculations indicate a longer discharge time at a higher discharge. However, the total discharge time of a single cycle system such as t' is unlikely to be long enough to thoroughly exercise the test valve.

8. The computer program used for the rotational system analysis is presented in Appendix C.

9. Appendix D presents the calculations upon which the dynamic analysis is based. It also outlines the program logic used in modeling the slowdown of the drive system.
PART III. IDEALIZED ANALYSIS, DROP TEST

Drop Test System

10. For comparison purposes, a swinging ball test fixture was evaluated. The system consisted of a ball suspended by a sling from a fixed point. When directly below the suspension point, the ball will impact the piston of a hydraulic cylinder connected to the regulating valve. Energy is stored in the ball by raising it to height \( H \) as shown in Figure 2.

![Diagram showing the drop test system with labels for key components:
- \( P \): Suspension Point
- \( H \): Drop Height in ft
- \( D \): Cylinder Diameter in in
- \( W \): Weight of the Ball in lbs
- Discharge Pressure: 4350 PSI

FIGURE 2

6
Dynamic Analysis, Drop

11. The dynamic analysis calculations and the computer program, shown in Appendix E, are based on total transfer of momentum from the ball to the pump system, i.e., the ball does not rebound. The results show the following:

a. The drop height controls the peak flow rate. Consequently, for a given cylinder diameter, H is fixed in order to attain the valve's design flow.

b. For a given cylinder diameter (thus drop height) increasing the weight of the ball increases the discharge time of the cylinder.

c. A 3.5 inch diameter cylinder, a 43.1 foot drop height, and a 2,000 lb ball will produce the design peak flow followed by a linear decline to zero flow for 78 msec.

d. Figure 3 is a composite plot showing the discharge versus time graphs of the best rotational test (from Figure B3) and the best drop test. This plot clearly shows the superior performance of the rotational system.

e. Time-discharge graphs of other cylinder diameters and ball weights are shown in Figure E2.
PART IV: CONCLUSIONS

12. The following conclusions are based on the idealized analysis of the two systems.

   a. Provided the initial rpm of the rotational test is increased from 424 to 577, the Drop Test will do an inferior job of exercising the regulating valve because of the Drop Test's linear decline in discharge. However, the drop test will cost less.

   b. The drop test is near the practical limits of ball weight and drop height. Going beyond the design discharge requires a four fold increase in drop height to double the peak discharge. Changing to a larger cylinder diameter increases discharge directly with the square of D, but decreases flow time with D$^2$.

   c. The very short test time of the rotating system produces extremely large forces in the bearings as well as other parts of the mechanism.

   d. The ability to increase rpm allows the rotational system to test at high discharges, provided the fixture can handle the peak forces. Consequently, it is more flexible.

   e. Precise timing for the start of the test is critical to getting meaningful results.

   f. Neither system provides enough time at high discharge to thoroughly test the regulator performance.
APPENDIX A: ROTATIONAL TEST RESULTS

1. The figures presented in this appendix show the rotational testing system's performance at the design shaft speed of 424 rpm. Other analysis (not presented) showed that the design discharge (26.4 gps) would be attained only when pumping action starts at a shaft angle of ninety degrees. This analysis shows the system's performance under those conditions.

2. Figure A1 provides a tabulation of various system parameters versus time. Figures A2 through A6 are plots of those same parameters versus time.
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PUMPING ENDS AT 0.071 SEC.
PUMP CUTOUT ANGLE = 70.8 DEGREES
NUMBER OF TIME STEPS = 1591
FIGURE A3

PROGRAM ROSSCOBRA/JPUMP - JAY CHEEK, NSWC, SP, WES, 18 JAN 1986
PISTON DIAMETER = 16.00 IN
SHAFT DIAMETER = 6.00 IN
PUMP PRESSURE = 4350.00 PSI
FLYWHEEL DIAMETER = 30.00 IN
FLYWHEEL WEIGHT = 2500.00 LB
CRANK THROW = 1.00 IN
CONNECTING ROD LENGTH = 30.00 IN
INITIAL RPM = 424.80
CRANK ANGLE AT START = 90.00 DEG

-50000 -45000 -40000 -35000 -30000 -25000 -20000 -15000 -10000 -50000

-50000 -40000 -30000 -20000 -10000 0

0 10 20 30 40 50 60 70 80

TIME IN MILI-SEC

TORMUE IN FT-LB
Program ROSSCOBRA/PUMP --- JAY CHEEK, SND. SL. WES. 16 JAN 1986
PISTON DIAMETER = 16.00 IN
SHAFT DIAMETER = 9.00 IN
PUMP PRESSURE = 4350.00 PSI
FLYWHEEL DIAMETER = 30.00 IN
FLYWHEEL WEIGHT = 2000.00 LB
CRANK THROW = 1.00 IN
CONNECTING ROD LENGTH = 30.00 IN
INITIAL RPM = 424.00
CRANK ANGLE AT START = 90.00 DEG

Figure A4
FIGURE A5

A6
APPENDIX B: ROTATIONAL TEST RESULTS, PUMP START AT ZERO DEGREES

1. The figures presented in this appendix show the rotational testing system's performance at two shaft speeds. Other analysis (not presented) shows that the design discharge (26.4 gps) would be attained at the design rpm (424) only when pumping action started at a shaft angle of ninety degrees. However, at that point the piston and fluid are moving at the maximum velocity making pump starting very difficult under those conditions. This analysis shows two sets of results for starting the test at zero degrees.

2. Figure B1 provides a tabulation of various system parameters versus time. That output shows the peak discharge reduced to about 16 gps when the shaft speed at pump start is 424 rpm. Figures B2 through B6 show plots of discharge and other system parameters versus time for the 424 rpm test.

3. Figures B7 and B8 show the peak system discharge to be at the design discharge when the shaft speed at pump start is increased to 577 rpm. Figures B9 through B12 show other system parameters versus time at 577 rpm.
## TEST START ANGLE, RPM = ?

- \( \theta = 0.424 \)

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**Pumping ends at 0.058 sec.**

**Pump cutout angle = 88.7 degrees**

**Number of time steps = 1303**

---

**FIGURE B1**

**B2**
Program ROSSCOBRA/PUMP -- JAY CHEEK, SMU, SL UES, 16 JAN 1966
Piston Diameter = 16.00 IN
Shaft Diameter = 9.00 IN
Pump Pressure = 4350.00 PSI
Flywheel Diameter = 30.00 IN
Flywheel Weight = 2000.00 LB
Crank Thru = 1.00 IN
Connecting Rod Length = 30.00 IN
Initial RPM = 424.00
Crank Angle at Start = 0.00 DEG
PROGRAM ROSSCOBRA/JPUMP -- JAY CHEEK, SMD, SL, WES, 16 JAN 1986
PISTON DIAMETER = 16.00 IN
SHAFT DIAMETER = 9.00 IN
PUMP PRESSURE = 4350.00 PSI
FLYWHEEL DIAMETER = 36.00 IN
FLYWHEEL WEIGHT = 2000.00 LB
CRANK THROU = 1.00 IN
CONNECTING ROD LENGTH = 30.00 IN
INITIAL RPM = 424.00
CRANK ANGLE AT START = 0.00 DEG

FIGURE B4
TEST START ANGLE, RPM =?

\[ \theta = 577 \]

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</tr>
<tr>
<td>900</td>
<td>0.0544</td>
<td>130.09</td>
<td>164.2</td>
<td>7177.</td>
<td>8.02</td>
<td>-38934</td>
</tr>
<tr>
<td>1000</td>
<td>0.0604</td>
<td>135.22</td>
<td>119.8</td>
<td>3817.</td>
<td>5.40</td>
<td>-35925</td>
</tr>
<tr>
<td>1100</td>
<td>0.0665</td>
<td>138.80</td>
<td>78.5</td>
<td>1639.</td>
<td>3.31</td>
<td>-33642</td>
</tr>
<tr>
<td>1200</td>
<td>0.0725</td>
<td>140.92</td>
<td>39.4</td>
<td>413.</td>
<td>1.59</td>
<td>-32219</td>
</tr>
<tr>
<td>1300</td>
<td>0.0786</td>
<td>141.66</td>
<td>1.5</td>
<td>1.0</td>
<td>0.06</td>
<td>-31718</td>
</tr>
</tbody>
</table>

PUMPING ENDS AT 0.079 SEC.
PUMP CUTOUT ANGLE = 141.7 DEGREES
NUMBER OF TIME STEPS = 1304

FIGURE B7

B8
FIGURE B9
APPENDIX C: ROTATIONAL SYSTEM PROGRAM "JPUMP"

1. This appendix presents a listing of program JPUMP that was used for the idealized dynamic analysis of the rotational test system. This program produced the tabulations and plots presented in Appendixes A and B.
TITLE FILED IN ROSSCOBRA/JFUMP
PUMP RUN-TIME TEST PROGRAM
JAY CHEEK, SMD, SL, WES, 14 JAN 1986

DP
FISTON DIAMETER (IN.)

DS
FUMP SHAFT DIAMETER (IN.)

AP
EFFECTIVE AREA OF PISTON (FT**2)

SP
STROKE OF THE FISTON (IN)

DF
FLYWHEEL DIAMETER (IN.)

WF
FLYWHEEL WEIGHT (LBS.)

PI
3.1415927

DTR
FACTOR TO CONVERT DEGREES TO RADIANS

G
32.2

P
WORKING PRESSURE IN THE PUMP (P/IN**2)

FM
FLYWHEEL MASS

RMI
ROTATIONAL MOMENT OF THE FLYWHEEL

RPM
INITIAL REV. PER MIN. OF THE FLYWHEEL

AVF
CURRENT ANGULAR VELOCITY OF THE FLYWHEEL (RAD./SEC)

AVFI
INITIAL ANGULAR VELOCITY OF THE FLYWHEEL (RAD./SEC)

EK
KINETIC ENERGY OF THE FLYWHEEL

THETA
SHFT ANGLE WHEN PUMPING TEST STARTS (DEG)

THETA
CURRENT SHAFT ANGLE DURING PUMP TEST

ZHI
INITIAL MOMENTUM OF THE FLYWHEEL

ZM
CURRENT MOMENTUM OF THE FLYWHEEL

ZIMP
TOTAL IMPULSE ON FLYWHEEL FROM PUMP ACTION

TDEL
TIME STEP (SEC)

TI
CURRENT TIME SINCE START OF THE TEST.

CL
CONNECTING ROD LENGTH.

TOR
TORQUE APPLIED DURING ONE TIME STEP

ALPHA
ANGULAR DECELERATION DURING ONE TIME STEP.

Q
DISCHARGE OF THE PUMP IN GAL/SEC.

XF
DIST. FROM CL OF CRANK TO PUMP AT STEP N

XFL
DIST. FROM CL OF CRANK TO PUMP AT STEP N-1

V
DIMENSION V(1500,6)

DATA PI/3.1415927/, G/32.2/, DP/16.0/, DS/9.0/

& DTR/57.295780/, WF/1000.0/, SP/2.0/, P/4350.0/

& DF/30.0/, CL/30.0/

CALL USTART

CALL UPSET('SPEED', 120)

CALL USET('SMALL')

CALL USET('X BOTHLABELS')

CALL USET('Y BOTHLABELS')

CALL UERASE

CALL UHOME

CALL UALPHA
INITIAL CONDITIONS

WRITE (6,110) ' TEST START ANGLE, RPM =?', THETA, RPM
READ (5,110) THETA, RPM

PISTON AREA
AP = PI * (DP * DP - DS * DS) / 576.0

MASS OF THE TWO FLYWHEELS
FM = WF * 2.0 / G

ROTATIONAL MOMENT OF INERTIA
RMI = FM * DF * DF / 1152.0

INITIAL ANGULAR VELOCITY
AVFI = RPM * PI / 30.0

CALC CONSTANTS FOR THE INTEGRATION LOOP.
CRANK ARM LENGTH IN FT
TH = SP / 24.0

SQUARE OF THE CONNECTING ROD LENGTH
CLS = CL * CL / 144.0

TIME STEP IN SEC.
TDEL = AVFI / 1000000.0

PUMP FORCE TIMES CRANK ARM LENGTH
FR = AP * F * TH * 144.0

STEP COUNTER
N = 0

INITIAL ANGULAR VELOCITY OF THE CRANKSHAFT (RADIANS/SEC)
AVF = AVFI

CRANKSHAFT ANGLE AT START OF PUMPING (0 DEG IS TOP DEAD CENTER OF STROKE)
THETA = THETA

NUMBER OF POINTS TO BE PLOTTED
NVP = 0

HEADING.
WRITE (6,120)
120 FORMAT ( ' N T(SEC) THETA RPM KE (FT-LB) Q ' , '(GAL/SEC) TORQ (FT-LB)')

START THE PUMP TEST LOOP

ELAPSED TIME IN SEC
130 TI = N * TDEL

CURRENT CRANKSHAFT ANGLE IN RADIANS
TR = THETA / DTR

KINETIC ENERGY IN FT LBS
EK = RMI * AVF * AVF / 2.0

TEST FOR SPECIAL CALCULATION FOR Q AT START
IF (N NE. 0) GO TO 140

CALC FUMP POSITION JUST BEFORE THE FIRST STEP.

TRR = TR - AVF * TDEL
SIN(TRR)

TEMP1 = TH * COS(TRR)
JPU M
1010 TEMP2 = SQRT(CLS-(TH*ST)**2)
1020 XPL = TEMP2 - TEMP1
1030 140 ST = SIN(TR)
1040 X COMPONENT OF THE CRANK ARM
1050 TEMP1 = TH * COS(TR)
1060 PROJECTION OF CONN, ROD ON X AXIS
1070 TEMP2 = SQRT(CLS-(TH*ST)**2)
1080 POSITION OF THE PUMP AT STEP N
1090 XP = TEMP2 - TEMP1
1100 PUMP DISCHARGE DURING THIS STEP
1110 Q = ABS(XP-XPL) * AP * 7.5 / TDEL
1120 XPL = XP
1130 SLOWDOWN TORQUE ON CRANK DUE TO PUMPING FORCE
1140 TOR = -ABS(FR*ST*(1.0-TEMPI/TEMP2))
1150 ANGULAR ACCELERATION
1160 ALFA = TOR / RMI
1170 IF (MOD(N,100) .EQ. 0) WRITE (6,150) N, TI, THETA, AVF /
1180 & 2. / PI * 60, EK, Q, TOR
1200
1210 SAVE EVERY FIFTH POINT FOR PLOTTING
1220 IF (MOD(N,5) .NE. 0) GO TO 160
1230 NVP = NVP + 1
1240 V(NVP,1) = TI * 1000.0
1250 V(NVP,2) = Q
1260 V(NVP,3) = TOR
1270 V(NVP,4) = EK
1280 V(NVP,5) = THETA
1290 V(NVP,6) = AVF / 2.0 / PI * 60.0
1300 NEW ANGULAR VELOCITY DUE TO SLOWDOWN TORQUE
1310 160 AVF = AVF + ALFA * TDEL
1320 COUNT STEF
1330 N = N + 1
1340 STILL ROTATING?
1350 IF (AVF .LT. 0.0) GO TO 170
1360 YES, ADVANCE TO NEXT CRANK ANGLE
1370 THETA = THETA + AVF * TDEL * DTR
1380 GO TO 130
1390
1400 DONE
1410 170 WRITE (6,180) TI, THETA - THETAI, N
1420 180 FORMAT (' PUMPING ENDS AT ', F10.3, ' SEC.', /
1430 & ' PUMP CUTOUT ANGLE = ', F8.1, ' DEGREES', /
1440 & ' NUMBER OF TIME STEPS = ', I5 // // )
1450 CALL UPAUSE
1460
1470 PLOT THE CURVES
1480 CALL PLOTFU(V, NVP, DP, DS, P, DF, WF, SP, CL, RPM, THETAI)
1490 GO TO 100
1500 END
SUBROUTINE PLOTPU(V, NV, DF, DS, P, DF, WF, SF, CL, RPM, \THETAI)

PLOT ANY OF THE 5 ARRAYS AS A FUNCTION OF TIME (ARRAY1).

JAY CHEEK, SMD, SL, WES, 16 JAN 1986

CHARACTER*20 T(20)
DIMENSION V(1500,6)
DATA T(1)/'TIME IN MILI-SEC'/, T(2) /'Q IN GAL PER SEC'/, T(3) /'TORQUE IN FT LB'/, T(4) /'KE IN FT LB'/, T(5) /'CRANK ANGLE IN DEG'/, T(6) /'REV PER MIN'/

START PLOTTING
FN = NV
CALL UPSET('XLABEL', T(1))
DO 120 I = 2, 6

IS THIS GRAPH NEEDED?
WRITE (6,100) T(I)
100 FORMAT ('PLOT', A20, '?')
CALL IANSR(IST)
IF (IST .EQ. 0) GO TO 120

YES, DO IT.
CALL UPSET('YLABEL', T(I))
CALL UERASE
CALL UDAREA(4.0, 14.0, 0.0, 10.0)
CALL USET('GRIDAXIS')
CALL UPLLOT(V(1, 1), V(1, I), FN)
CALL UHOME
CALL UALPHA
WRITE (6,110) DF, DS, P, DF, WF * 2.0, SF / 2.0, CL, RPM, \THETAI
110 FORMAT ('PROGRAM ROSSCOBRA/JPUMP -- JAY CHEEK, SMD, ','SL, WES, 16 JAN 1986' / 'PISTON DIAMETER =', 'FB,2, ' IN' / 'SHAFT DIAMETER =', 'FB,2, ' IN' / 'PUMP PRESSURE =', 'FB,2, ' PSI' / 'FLYWHEEL DIAMETER =', 'FB,2, ' IN' / 'FLYWHEEL WEIGHT =', 'FB,2, ' LB' / 'CRANK THROW =', 'FB,2, ' IN' / 'CONNECTING ROD LENGTH =', 'FB,2, ' IN' / 'INITIAL RPM =', 'FB,2, ' RPM' / 'CRANK ANGLE AT START =', 'FB,2, ' DEG')
CALL UPAUSE
120 CONTINUE
RETURN
END
SUBROUTINE IANSR(IWHAT)

SEE WHETHER THE USER GIVES A Y (OR A CARRIAGE RETURN) FOR YES OR A N FOR NO TO A PREVIOUSLY ASKED QUESTION.

CODE FILED IN ROSSCOBRA/JHEST-S.
JAY CHEEK, SMD, SL, WES; DEC 1981

DATA IBLK/' /, IYES/'Y' /, NO/'N' /
100 IWHAT = 0
READ (5,110) II
110 FORMAT (A4)
120 FORMAT (A4)
130 IF (II .EQ. NO) RETURN
140 IF (II .EQ. IYES) RETURN
150 IF (II .EQ. IBLK) RETURN
160 WRITE (6,120)
170 120 FORMAT (' ERROR: ONLY Y OR RETURN (FOR YES) OR N (FOR N) ALLOWED, RETRY')
180 GO TO 100
190 END
APPENDIX D: METHOD FOR CALCULATING SLOWDOWN OF FLYWHEEL

Initial Conditions

1. As shown in Figure D1, a flywheel (solid disc) of weight W is rotating at an initial rpm. The flywheel diameter is D. The flywheel is connected to a crankshaft whose offset (throw) is T. A connecting rod of length C connects the crank to the pump that resists motion with a constant force F. That force is directed on a line from the crankshaft center-line through the center-line of the pump shaft.

Pump's End Plates and Valves not shown

Idealized Pump and Drive System

Figure D1
Calculating Procedure For Flywheel Slowdown

2. The following steps are used to calculate the flywheel slowdown:
   a. Calculate the Rotational Moment of Inertia (I).
      \[ I = \frac{m D^2 \left( \frac{D}{2} \right)^2}{2} \]

   b. Calculate the Angular Velocity (\( \omega \)).
      \[ \omega = \frac{2\pi (\text{rpm})}{60} \]
      and choose a small time step (\( \Delta t \)) so that the wheel will rotate less than .1° at the initial rpm during time \( \Delta t \).

   c. At each time Step i, Calculate:
      1. The angle traversed (\( \Delta \theta \)) during this time step (constant angular velocity (\( \omega \)) is assumed).
         \[ \Delta \theta = \omega_i \Delta t \]
      2. The current angular location (\( \theta \)) of the crank.
         \[ \theta_i = \theta_{i-1} + \Delta \theta \]
      3. The component of \( F \), acting through the connecting rod that is perpendicular to the crank arm (\( F_M \)).
      4. The torque (\( \Gamma \)) at angle \( \theta \).
         \[ \Gamma = F_M T \]
      5. The angular deceleration (\( \alpha \)) produced by this constant torque.
         \[ \alpha = -\frac{\Gamma}{I} \]
6. The new angular velocity due to the slowdown torque from the pumping force.

\[ \omega_i = \omega_i - \alpha \Delta t \]

7. Add 1 to i and repeat steps C1 through C7 until \( \omega_i \leq 0 \).

8. Done.

Calculating Moment Applied To The Crankshaft

3. As illustrated in Figure D2: \( T \) is crankshaft throw, \( F \) is force to operate the pump, \( C \) is connecting rod length, \( E \) is the moment arm, \( \theta \) is crankshaft angle, and \( G \) is the force normal to \( E \).

![Figure D2](image)

Calculating the Torque (\( \Gamma \)) about point \( Q \).

\[
\begin{align*}
\Gamma &= GE \\
E &= A \sin \phi \\
A &= B - T_X \\
B &= (C^2 - T_Y^2)^{1/2} \\
T_Y &= T \sin \theta \\
T_X &= T \cos \theta \\
G &= \frac{F}{\cos \phi} \\
\cos \phi &= \frac{B}{C} \\
\Gamma &= \frac{F}{B} (B - T_X) \frac{T_Y}{C} = FT_y (1 - \frac{T_X}{B}) \\
&= FT \sin \theta \left[ 1 - \frac{T \cos \theta}{(C^2 - T^2 \sin^2 \theta)^{1/2}} \right] \text{ for } C > T
\end{align*}
\]
APPENDIX E: ANALYSIS OF THE DROP TEST SYSTEM

1. This appendix contains the output of the drop test analysis program, JPEND (Figure E1), the listing of the program (E3), and the development of the equations used (E4 and E5). The output is given for both 1,000 and 2,000 pound balls impacting on the pistons of various diameter cylinders. The drop height is paired with the cylinder diameter to produce the rated discharge peak of 26.4 gps. Discharge pressure is assumed to be constant at 4,350 psi. Note that an increase in ball weight, holding other parameters constant, serves to increase the test time (t) directly as the ratio of the new weight to the old

\[ t_N = \frac{W_N}{W_0} t_0 \]

where subscripts 0 and N refer to old and new values.

Figure E2 gives the discharge versus time curves for several ball weight, cylinder diameter and drop height combinations.
Output of the Drop Test Analysis Program "JPEND"

**DROP TEST OF VALVE**
**FOR BALL WEIGHT OF 1000, LB PEAK DISCHARGE OF 26.4 GAL / SEC AND DISCHARGE PRESSURE OF 4350, PSI**

<table>
<thead>
<tr>
<th>CYLINDER DIAMETER (INCHES)</th>
<th>DROP HEIGHT (FEET)</th>
<th>RETARD FORCE (POUNDS)</th>
<th>IMPACT VELOCITY (FT / SEC)</th>
<th>TIME (SECONDS)</th>
<th>DISCHARGE ENERGY (FT - LBS)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.50</td>
<td>43.10</td>
<td>41852.</td>
<td>52.68</td>
<td>0.039</td>
<td>43101.</td>
</tr>
<tr>
<td>3.75</td>
<td>32.71</td>
<td>48044.</td>
<td>45.89</td>
<td>0.030</td>
<td>32706.</td>
</tr>
<tr>
<td>4.00</td>
<td>25.26</td>
<td>54664.</td>
<td>40.34</td>
<td>0.023</td>
<td>25265.</td>
</tr>
<tr>
<td>4.25</td>
<td>19.82</td>
<td>61710.</td>
<td>35.73</td>
<td>0.018</td>
<td>19824.</td>
</tr>
<tr>
<td>4.50</td>
<td>15.77</td>
<td>69184.</td>
<td>31.87</td>
<td>0.014</td>
<td>15773.</td>
</tr>
<tr>
<td>4.75</td>
<td>12.71</td>
<td>77084.</td>
<td>28.60</td>
<td>0.012</td>
<td>12705.</td>
</tr>
<tr>
<td>5.00</td>
<td>10.35</td>
<td>85412.</td>
<td>25.82</td>
<td>0.009</td>
<td>10348.</td>
</tr>
</tbody>
</table>

**DROP TEST OF VALVE**
**FOR BALL WEIGHT OF 2000, LB PEAK DISCHARGE OF 26.4 GAL / SEC AND DISCHARGE PRESSURE OF 4350, PSI**

<table>
<thead>
<tr>
<th>CYLINDER DIAMETER (INCHES)</th>
<th>DROP HEIGHT (FEET)</th>
<th>RETARD FORCE (POUNDS)</th>
<th>IMPACT VELOCITY (FT / SEC)</th>
<th>TIME (SECONDS)</th>
<th>DISCHARGE ENERGY (FT - LBS)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.50</td>
<td>43.10</td>
<td>41852.</td>
<td>52.68</td>
<td>0.078</td>
<td>86201.</td>
</tr>
<tr>
<td>3.75</td>
<td>32.71</td>
<td>48044.</td>
<td>45.89</td>
<td>0.059</td>
<td>65413.</td>
</tr>
<tr>
<td>4.00</td>
<td>25.26</td>
<td>54664.</td>
<td>40.34</td>
<td>0.046</td>
<td>50530.</td>
</tr>
<tr>
<td>4.25</td>
<td>19.82</td>
<td>61710.</td>
<td>35.73</td>
<td>0.036</td>
<td>39649.</td>
</tr>
<tr>
<td>4.50</td>
<td>15.77</td>
<td>69184.</td>
<td>31.87</td>
<td>0.029</td>
<td>31545.</td>
</tr>
<tr>
<td>4.75</td>
<td>12.71</td>
<td>77084.</td>
<td>28.60</td>
<td>0.023</td>
<td>25410.</td>
</tr>
<tr>
<td>5.00</td>
<td>10.35</td>
<td>85412.</td>
<td>25.82</td>
<td>0.019</td>
<td>20697.</td>
</tr>
</tbody>
</table>

FIGURE E2
Listing of the Drop Test Analysis Program "JPEND"

JPEND
10C$TITLE FILED IN ROSSCOBRA/JPEND
20C CALC OF PENDULUM TEST OF VALVE PERFORMANCE.
30C JAY CHEEK SMD, SL, WES, 17 JAN 1986
40C
50 DATA PI/3.1415926/, P/4350.0/, Q/26.4/
60C
70 DO 130 J = 1, 2
80C WEIGHT OF BALL
90 WT = J * 1000.0
100C DIAMETER OF HYDRAULIC CYLINDER (ONE STROKE PUMP)
110 D = 3.250
120 WRITE (6,100) WT, Q, P
130 100 FORMAT (///, 'DROP TEST OF VALVE' /
140 & ' FOR BALL WEIGHT OF', F6.0, 'LB', ' PEAK DISCHARGE OF', F5.1, 'GAL / SEC' /
150 & ' AND DISCHARGE PRESSURE OF', F6.0, 'PSI' /
160 & ' CYLINDER DROP RETARD IMPACT', 'DIAMETER HEIGHT FORCE VELOCITY TIME' /
170 & ' DISCHARGE KINETIC' /
180 & ' DIAMETER HEIGHT FORCE VELOCITY TIME' /
190 & '(INCHES) (FEET) (POUNDS) (FT / SEC)' /
200 & '(SECONDS) (FT - LBS)' /
210 & 'CALC FOR SEVERAL CYL. DIAMETERS'
220 DO 120 I = 1, 7
230C DIAMETER
240D DI = D + I * .25
250C PISTON (CYLINDER) AREA
260A = PI * DI * DI / 4.0
270C MASS OF THE BALL
280ZMASS = WT / 32.2
290C DROP HEIGHT TO PRODUCE PEAK DISCHARGE
300H = 9.230 * Q * Q / DI ** 4
310C IMPACT VELOCITY
320V = SQRT(2.0*32.2*H)
330C RETARDING FORCE OF THE CYLINDER
340F = P * A
350C TIME OF DISCHARGE
360TI = ZMASS * V / F
370C KINETIC ENERGY
380EK = ZMASS * V * V / 2.0
390C RESULTS
400WRITE (6,110) DI, H, F, V, TI, EK
410110 FORMAT (1X, F7.2, F10.2, F12.0, F10.2, F10.3, F10.0)
420120 CONTINUE
430130 CONTINUE
440STOP
450END

E3
Equation For Drop Height

2. Calculating combinations of cylinder diameter (D) and drop height (H) that yield a specific value of fluid discharge Q.

a. Impact velocity \( U_I = \sqrt{2gH} \)

where \( U \) is in ft/sec

\( H \) is in ft

\( g \) is in \( \text{ft/sec}^2 \)

b. Discharge \( Q = \frac{V}{\Delta t} \) where \( V \) is volume in \( \text{ft}^3 \).

\[
Q = \frac{V}{\Delta t} = \frac{A \Delta x}{\Delta t} = AU_I
\]

where \( A \) is cylinder area = \( \frac{\pi}{4} D^2 \)

\( D \) is cylinder diameter in ft

c. \( H = \frac{Q_I^2}{2g \left( \frac{\pi}{4} D^2 \right)^2} = \frac{Q_I^2}{1.234gD^4} \)

where \( Q_I \) is discharge at impact, i.e., peak discharge.

d. Converting for \( Q \) in gps, and \( D \) in inches.

\[
H = \frac{Q_I^2}{7.5 \left( \frac{D}{12g} \right)^4} = \frac{9.280 Q_I^2}{D^4}
\]
Equation For Total Discharge Time

3. As shown on E4, Drop Weight (H) relates to Peak Discharge (Q₁) and cylinder diameter (D) by:

\[ H = \frac{9.280Q^2}{D^4} \]

Where Q is in gps, D is in inches.

Conservation of momentum gives \( MU_1 = Ft \)

Where t is the time in seconds that the constant force F exists, M is the ball mass, W is the ball weight and \( M = \frac{W}{g} \).

\( F \) is the constant piston force = \( \frac{\pi D^2 P}{4} \)

Where P is pressure in psi, F is in lbs.

\( U_1 = \) impact velocity = \( \left(\frac{2gH}{2} \right)^{\frac{1}{2}} \)

Total discharge time (t) is

\[ t = \frac{MU_1}{F} = \frac{W}{g} \left(\frac{2gH}{2} \right)^{\frac{1}{2}} = \frac{W}{g} \left(\frac{2g9.280Q^2}{D^4} \right)^{\frac{1}{2}} = .9667 \frac{W}{P} \frac{Q^2}{D^4} \]

Where: W is weight in lb, D is diameter in inches, Q is discharge in gps, P is pressure in psi, and t is time in sec.

For \( P = 4,350 \) psi, \( Q₁ = 26.4 \) gps

\[ t = .005866 \frac{W}{D^4} \]
DROP TEST OF PRESSURE REGULATING VALVE

KEY:

P is the suspension point
H is drop height in ft
D is cylinder diameter in in
W is weight of the ball in lbs.
Discharge pressure = 4350 psi

Figure E2
END

DTIC

8-86