Introduction to *Team* Corporation Engine Simulation Systems

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Introduction

The *Team* 900 Series of Engine Simulation Systems (ES S) provide unique test capabilities to developers and manufacturers of engine driven components and systems. This document has been prepared to introduce the reader to the range of applications and theory of operation of the *Team* ESS.

Before discussing the details of the *Team* 901 Engine Simulation System it is useful to address two common questions:

What Does an Engine Simulator Do?

The obvious answer is that an engine simulator simulates an engine. Specifically, the engine simulator produces torsional vibrations similar to those found on the crankshaft of an engine. Like an engine, the engine simulator produces the torsional vibrations while spinning.

Why Use an Engine Simulator?

There are several advantages to using an engine simulator instead of an engine. Some of these are:

Consistency: Unlike an engine, the engine simulator is consistent and repeatable. The characteristics of the simulator do not change over time.

Programmability: The engine simulator can be programmed to simulate virtually any engine configuration. Any number of cylinders can be simulated easily.

Availability: In the early stages of an engine development program prototype engines are often unavailable. The engine simulator can be programmed to simulate the prototype engine.

Reduced Facility Costs: Unlike an engine, the engine simulator does not require fuel or exhaust gas handling systems. The resulting facility savings are often significant.

Flexibility: The engine simulator can be used to test front engine accessories, harmonic dampers, clutches, transmissions, and drivelines.

System Overview

The 901 Engine Simulation System may be divided into several primary sub-systems. These are:

Primary Sub-System	Function
Variable Speed Drive System (Optional)	Provides rotary motion and steady state torque for the engine simulation system. A variety of motor types and power ratings are available.
901 Spinning Rotary Actuator	Provides torsional vibrations to the test article
Instrumentation	Provides feedback of acceleration, position and torque to the Vibration Control System or an independent data acquisition system. Additional instrumentation may be installed for specific requirements.
Vibration Control System	Programs and controls the torsional vibration frequency and amplitude.
Absorber System (Optional)	Works with the Variable Speed Drive to provide steady state torque to the test article.

Table 1: Engine Simulation System - Primary Sub-Systems

The Engine Simulation System also includes the following supporting sub-systems:

Supporting Sub-System	Function
Hydraulic Power Supply	Provides hydraulic pressure and flow to the 901 Spinning Rotary Actuator.
Valve Driver Electronics Package	Amplifies the drive signal from the Vibration Control System and conditions the position signals from the 901 Spinning Rotary Actuator.

Table 2: Engine Simulation System - Supporting Sub-Systems

Figure 1 illustrates the relationship of the primary and supporting sub-systems.

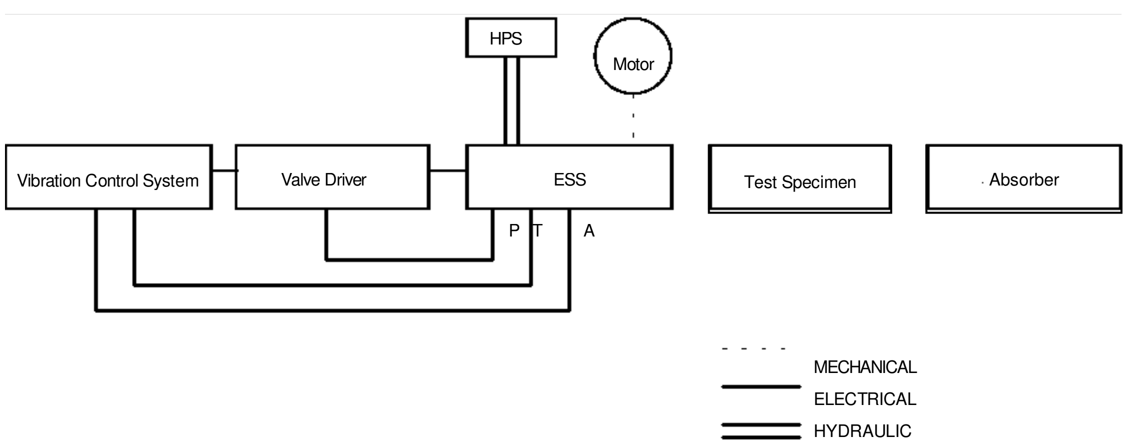

Figure 1 - Block Diagram - Typical Engine Simulation System.

The Spinning Rotary Actuator

A typical *Team* Spinning Rotary Actuator is shown in Figure 2. Please refer to this illustration for the following discussion.

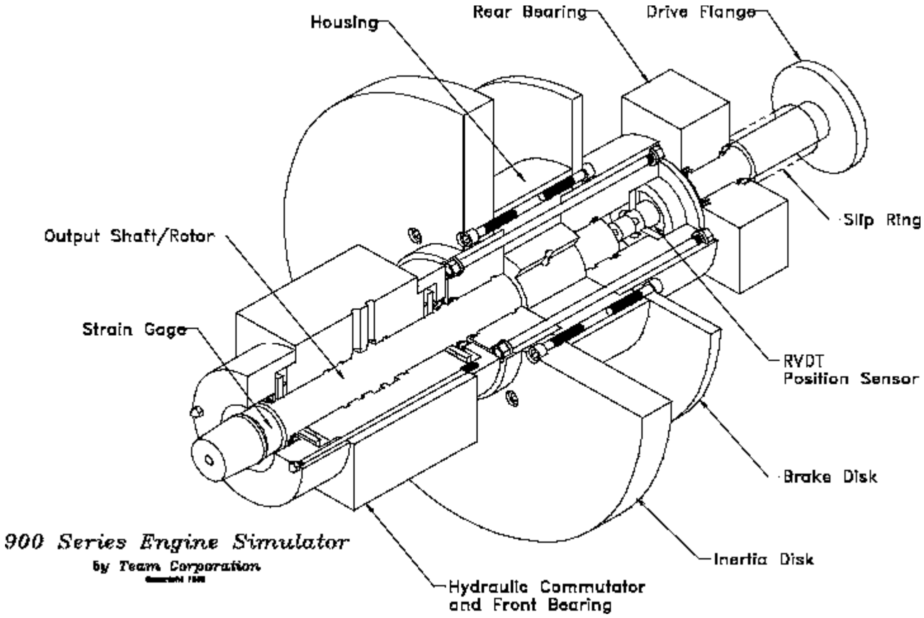

Figure 2 - Typical Team 900 Series Spinning Rotary Actuator.

The Spinning Rotary Actuator can best be described as a rotary actuator whose housing is allowed to rotate. All dynamic torque, acceleration and displacement is generated relative to the actuator housing. Steady state torque, produced by the prime mover and absorber, acts through the spinning rotary actuator.

The primary working components of the spinning rotary actuator are the housing and the rotor. The housing is driven by the prime mover. The rotor maintains its position relative to the housing due to the hydraulic pressure on the rotor vanes. Dynamic torque is produced by modulating the pressure on the rotor vanes.

The front bearing of the actuator assembly has two functions. It is a hydrostatic journal bearing supporting the actuator and allowing friction-free rotation. It is also a hydraulic slip ring allowing oil to flow into the actuator.

Attached to the housing are two disks. The larger disk is the inertia disk. The inertia disk reacts the dynamic torque generated by the rotor.

The second and smaller disk is a brake disk. The engine simulation system is equipped with a disk brake for use in an emergency. The brake can be triggered manually or automatically.

Servo Valve Assembly

(not shown)

A two stage high performance servo valve is mounted on top of the front bearing/hydraulic slip ring. The *Team* servo valve uses a voice coil driven pilot valve for high frequency operation. Please refer to the section *'Servohydraulic Fundamentals'* for more information on the operation of the servovalve assembly.

Support Frame

(Not shown)

The support frame is the structural foundation of the engine simulator. The support frame is constructed of welded steel tubing. The support frame may be designed to accommodate existing dynamometer facilities.

Input Flange

The input flange is the point of attachment for the prime mover. The flange is often designed for specific applications, so the details will vary from system to system.

Output Flange

The output flange is the point of attachment for the test specimen or system under test. The flange is often designed for specific applications, so the details will vary from system to system.

It is very important that the output flange not be overloaded. The maximum radial load at the flange is 200 lb. If the system under test exceeds this loading, additional support will be required. If the system under test includes bearings, extra care must be taken to align the system under test and the ESS to prevent damage to the ESS and the system under test.

Instrumentation

The Engine Simulator System includes instrumentation for measurement of acceleration and position. These signals may be used for measurement or control.

Accelerometer

The output flange of the Engine Simulation System accepts a Dytran Model 31 01A3 (or equivalent) accelerometer. The accelerometer is mounted tangential to the axis of rotation. This allows measurement of the angular acceleration of the output flange.

The accelerometer is the primary control feedback. It is used for both acceleration and position control. Position being derived by double integrating the acceleration signal.

RVDT

The Engine Simulation System includes a built in Rotary Variable Differential Transformer (RVDT) for measurement of the rotor position relative to the housing.

The RVDT is used for feedback to the position control servo loop included in the valve driver electronics.

Slip Ring

The Engine Simulation System includes an Michigan Scientific Model SR10-AA (or equivalent) slip ring assembly. This model has 10 rings and is rated at 100 million revolutions. The slip ring carries the acceleration, position and torque signals. Slip ring models with higher durability rating available upon request.

Performance Specifications

Team Corporation manufactures three models of Engine Simulation Systems. The following table provides the performance specifications for each of these models.

	***Team* Model 901**	***Team* Model 902.5**	***Team* Model 904**
Max. Total Torque	10,000 in-lb[1]	25,000 in-lb[1]	40,000 in-lb[1]
Angular Travel	60 degrees	60 degrees	60 degrees
Angular Velocity	25 radians/sec	25 radians/sec	15 radians/sec
Maximum Speed	6,000 rpm[2]	3,600 rpm	3,600 rpm
Shaft Inertia	0.1 lb-in-sec^2	0.8 lb-in-sec^2	0.8 lb-in-sec^2
Reaction Inertia	11 lb-in-sec^2	150 lb-in-sec^2	150 lb-in-sec^2
Operating Temperature Range	65-110 °F	65-110 °F	65-110 °F

Table 3: Team ESS System Performance Specifications

As an example, the following chart illustrates the No-Load Performance Envelope of the *Team* 901 Engine Simulation System.

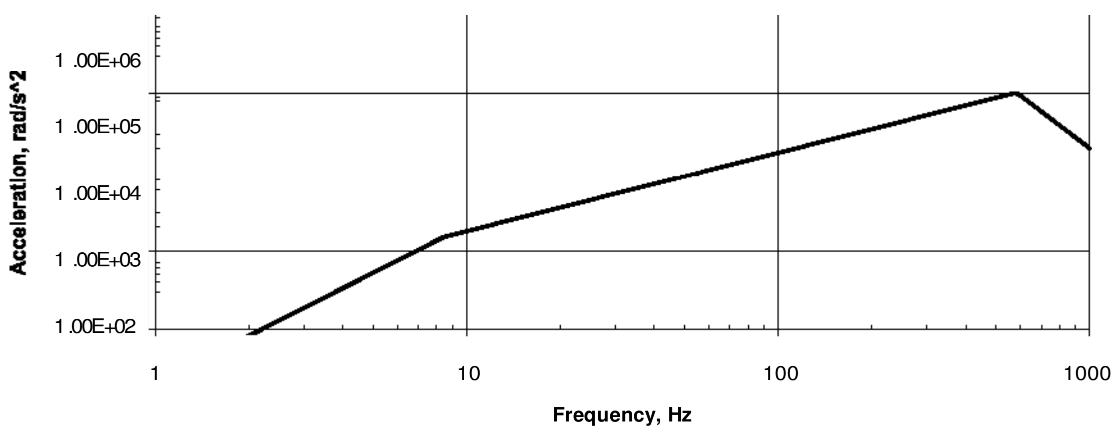

Figure 3 - No Load Performance of Team 901 Engine Simulation System.

[1] Note that any steady state torque acting through the spinning rotary actuator must be offset by hydraulic pressure on the rotor vanes. This means that some of the available hydraulic pressure must be used to offset any steady state torque. The result is a reduction of the available dynamic torque by an amount equal to the steady state torque.

[2] 10,000 rpm on some models.

Test Evaluation

Proper evaluation of test requirements will result in improved test performance and reduce the possibility of damage to the test system. Evaluating the test specification is a simple procedure that takes only a few moments, yet can save hours of frustration.

Sinusoidal Vibration Equations

All of the following discussion assumes that the vibration waveform is sinusoidal. For a sinusoidal waveform, the rotation values for angular acceleration, angular velocity, and angular displacement are given by the following general equations.

(1) $$\alpha = \alpha_0 \sin(2\pi f t)$$

(2) $$\omega = \frac{\alpha_0}{2\pi f} \cos(2\pi f t)$$

(3) $$\theta = \frac{\alpha_0}{(2\pi f)^2} \sin(2\pi f t)$$

Where:

α = Angular Acceleration, radians/sec^2

ω = Angular Velocity, radians/sec

θ = Angular Displacement, radians (0-Peak)

α_0 = Peak Angular Acceleration, adians/sec^2

π = Pi, 3.14159

f = Frequency, Hz

t = time, sec

The values of interest, the peak magnitudes, can be found by dropping the trigonometric functions and evaluating only the coefficients. These equations can be further manipulated to yield values for any two unknowns given any one known. The following table provides the various forms of the equations.

Known Value	**Unknown Values**		
	α	ω	θ
α		$\frac{\alpha_0}{2\pi f}$	$\frac{\alpha_0}{(2\pi f)^2}$
ω	$2\pi f\omega$		$\frac{\omega}{2\pi f}$
θ	$(2\pi f)^2\theta$	$\omega = 2\pi f\Theta$	

Table 4 - Angular Vibration Equations, Sinusoidal Motion

Newton's Law

The other important relationship is Newton's law, expressed in angular terms:

$$T = I\,\alpha \qquad (4)$$

Where:

T = Torque, in-lbs

I = Inertia, in-lbs-s^2

α = Angular Acceleration, radians/sec^2

In Equation (4), the inertia, I, is the total inertia at the output flange of the ESS. If the system under test includes multiple inertia with multiple pulley ratios, each inertia must be 'reflected' back to an equivalent inertia at the output flange and summed. This will give the worst case loading.

Evaluation Procedure

The process of evaluating the test specification proceeds as follows:

1. Using the equations in Table 4, compute values for peak angular acceleration, peak angular velocity and peak angular displacement.
2. Compute the total driven inertia of the system under test.
3. Using Equation (4), compute the peak dynamic torque required for the test.
4. Add any steady state torque to the peak dynamic torque to get the total torque.
5. Compare the values for the total torque, peak angular velocity and peak angular displacement to the maximum performance limits of the ESS. If all of the computed values are equal to or less than the performance limits of the ESS, the ESS will perform the test.

The following example will illustrate this process.

Example

Problem:

Simulate a 6-cylinder engine operating at idle (750 Rpm). The speed variation measured at the crank is ±40 Rpm. The following information is also known:

	Crankshaft	Water Pump	P/S	Alternator	A/C
Inertia, ft-lbm	5.00	6.00	4.49	9.34	14.10
Torque, ft-lbf	-	5.00	18.00	3.50	9.00
Pulley Ratio	1.00	1.29	1.22	2.82	1.47

Solution:

Step 1: Compute the peak angular acceleration, velocity and displacement.

First, we need to know the peak angular acceleration, velocity and displacement of the crank. To find this we need to convert the speed variation into more convenient unite, namely, rad/s.

$$\frac{40\ rev}{min} \times \frac{2\tau\ rad}{rev} \div \frac{60\ sec}{min} = 4.19\ \frac{rad}{sec}$$

This is the peak angular velocity.

Next, calculate the frequency of the torsional pulsation. For any 4-cycle engine, the following formula applies:

$$f = \frac{\#\text{ of cylinders X rpm}}{120}$$

Making the appropriate substitutions for our case yields:

$$f = \frac{6 \text{ X } 750}{120} = 37.5 \text{ Hz}$$

Now, apply the formulas from Table X to calculate the peak angular acceleration and displacement.

$$\alpha = 2\pi f \omega = 2\pi(37.5)(4.19) = 987.2 \frac{rad}{sec^2}$$

$$\theta = \frac{\omega}{2\pi f} = \frac{4.19}{2\pi(37.5)} = 0.018 \text{ rad}$$

Step 2: Compute the total driven inertia.

All of the accessory inertia must be 'reflected' back to the crankshaft. Use the pulley ratios to calculate the equivalent inertia. The table below shows the results.

	Inertia, ft-lbm	Ratio		Equivalent Inertia
Crankshaft	5.00	$(1/1.00)^2$	=	5.00
Water Pump	6.00	$(1/1.29)^2$	=	9.98
P/S	4.49	$(1/1.22)^2$	=	6.68
Alternator	9.34	$(1/2.82)^2$	=	74.28
A/C	14.10	$(1/1.47)^2$	=	30.47
TOTAL				126.41
Convert to in-lb-s^2			X	0.0311
Total Driven Inertia				**3.93 in-lb-s^2**

Step 3: Compute the Peak Dynamic Torque.

Dynamic torque is computed using the total driven inertia and the angular acceleration. The formula is Newton's Law, equation 4.

$$T_D = I_{total}\,\alpha$$

Substituting the values for our case yields: T_D

= (3.93) (987.2) = 3881 in-lbf Step 4: Add

any steady torque.

Again, use the pulley ratios to calculate the reflected torque of each accessory at the crank. The table below gives the results.

	Ts, ft-lbf		Ratio		Equivalent Ts
Crankshaft	-	x	1.00	=	-
Water Pump	5.00	x	1.29	=	6.45
P/S	18.00	x	1.22	=	21.96
Alternator	3.50	x	2.82	=	9.87

A/C	9.00	x	1.47	=	13.23
TOTAL					51.50
Convert to in-lbf				X	12
Total Steady Torque					**618 in-lbf**

Adding the dynamic and steady torque to get the Total Torque:

$$T_{otal} = T_D + T_S = 3881 + 618 = 4499 \text{ in-lbf}$$

Step 5: Compare the results to the performance specifications of the ESS.

All of the computed values are within the range of the Model 901 ESS. Therefore there is high level of confidence that the simulation will be successful.

Servohydraulic Fundamentals

The following section is excerpted from *'Shock and Vibration Handbook'*, Second Edition, edited by Cyril M. Harris and Charles E. Crede.

Hydraulic Vibration Machine

The *hydraulic vibration machine* is a device, which transforms power in the form of a high-pressure flow of fluid from a pump to a reciprocating motion of the table of the vibration machine. A schematic diagram of a typical machine is shown in Figure 4. In this example, a two stage electrohydraulic valve is used to deliver high-pressure fluid, first to one side of the piston in the actuator and then to the other side, forcing the actuator to move with a reciprocating motion. This valve consists of a pilot stage and power stage, the former being driven with a reciprocating motion by the electrodynamic driver. At the time the actuator moves under the force of high-pressure fluid on one side of the piston, the fluid on the other side of the piston is forced back through the valve at reduced pressure, and returned to the pump.

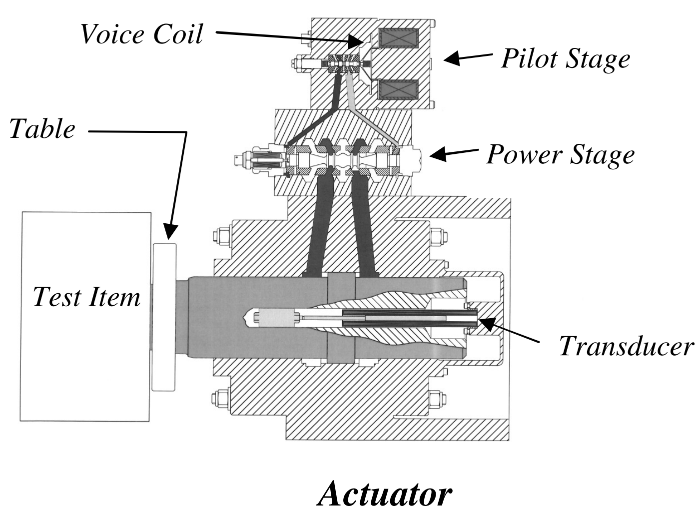

Figure 4 - Schematic Diagram of a typical hydraulic vibration machine

The electrohydraulic valve is usually mounted directly on the side of the actuator cylinder, forming a close-coupled assembly of massive steel parts. The proximity of the valve and cylinder is desirable in order to reduce the volume and length of the connecting fluid paths between the several spools and the actuator, thereby minimizing the effects of the compliance of the fluid and the friction to its flow.

Operating Principle

In Figure 4, the *pilot* and *power spools* of a hydraulic vibration machine are shown in the “middle” or “balanced” position, blocking both the pump high-pressure flow *P* and the return low-pressure flow *R*. Correspondingly, the piston of the actuator must be stationary since there can be no fluid flow either to or from the actuator cylinder. If the pilot spool is displaced to the right of center by a force from the electrodynamic driver, the high-pressure fluid *P* will flow through the passage from the pilot spool to the left end of the power spool, causing it to move to the right also. This movement forces the trapped fluid from the right-hand end of the power spool through the connecting passage, back tot he pilot stage, and then through the opening caused by the displacement of the pilot spool to the right, to the chamber *R* connected to the return to the pump. Correspondingly, if the pilot spool moves to the left, the flow to and from the power spool is reversed, causing it to move to the left. For a given displacement of the pilot spool, a flow results which causes a corresponding velocity of the power spool. A displacement of the power spool to the right allows the flow of high-pressure fluid *P* from the pump to the left side of the piston in the actuator, causing it to move to the right. This forces the trapped fluid on the right of the piston to be expelled through the connecting passage to the power spool and out past the right-hand restrictions to the return fluid chamber *R*. The transducers shown on the power spool and the actuator shaft are of the differential transformer type and are used in the feedback circuit to improve system operation and provide electrical control of the average (i.e., stationary) position of the actuator shaft relative to the actuator cylinder.

A block diagram of the complete hydraulic machine system is shown in Figure 5. The pump, in conjunction with accumulators in the pressure and return lines at the hydraulic valve, should be capable of variable flow while maintaining a fixed pressure. Most systems to date have required an operating pump pressure of 3,000 $lb.in.^2$ The upper limit of efficiency of the hydraulic valve is approximately 60 per cent, the losses being dissipated in the form of heat. Mechanical loads are seldom capable of dissipating appreciable power; most of the power in the pump discharge is converted to a temperature rise in the fluid. Therefore a heat exchanger limiting the fluid temperature must be included as part of the system.

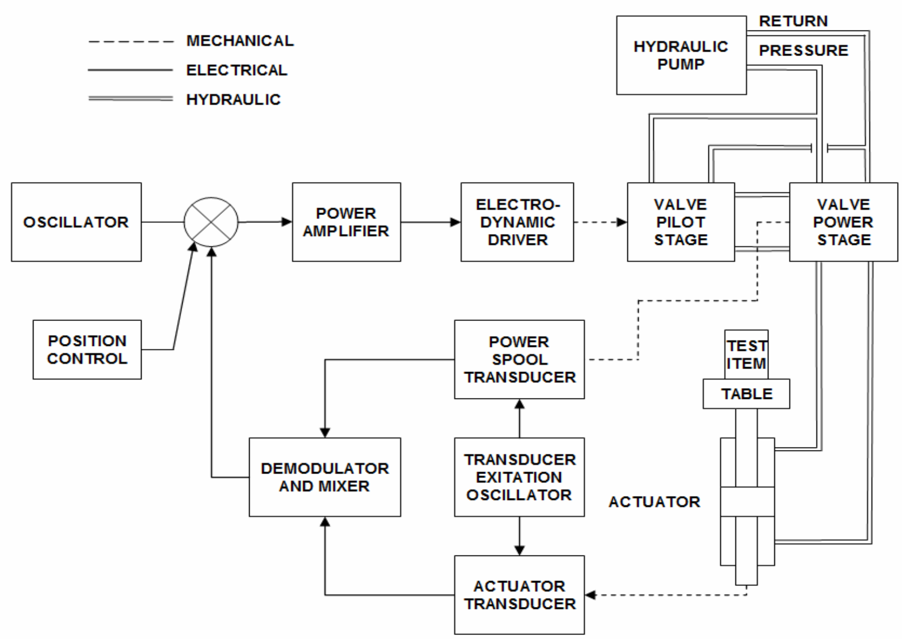

Figure 5 - Block diagram - hydraulic vibration machine system.