To the Graduate Council:

I am submitting herewith a thesis written by CPT(P) Christopher F. Smith, USA entitled "Application of Horsepower Absorbtion Techniques to Turboshaft Engine Testing." I have examined the final copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science,with a major in Aerospace Engineering.

Roy J. Schulz, Major Professor

We have read this thesis
and recommend its acceptance:

Accepted for the Council:

Associate Vice Chancellor and
Dean of the Graduate School

APPLICATION OF HORSEPOWER ABSORPTION TECHNIQUES TO TURBOSHAFT ENGINE TESTING

A Thesis

Presented for the

Master of Science

Degree

University of Tennessee, Knoxville

Christopher Francis Smith
CPT(P) United States Army
May 1996

DEDICATION

This thesis is dedicated to my wife

Krystal Rae Smith

without whom my life would be incomplete

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ABSTRACT

This investigation is an attempt to develop a standardized matrix for identifying the optimal horsepower absorber to utilize for a particular turboshaft engine test. Previous efforts identified optimal absorbers within a given family of devices for a particular facility or engine type. Herein, after a historical perspective, the several types of horsepower absorbers currently in industry-wide use are generalized with respect to their performance and operating tendencies. An in-depth analysis of each device's operating principles is coupled with an analysis of each device's positive and negative operating characteristics, thereby illustrating necessary trade-off factors. A discussion of current procedures for turboshaft engine testing is included to define the potential device use environment. An in-depth discussion and qualitative analysis of current turboshaft engine applications is carried out to provide an appreciation for the load modeling considerations necessary in turboshaft engine testing. Modeling requirements, device characteristics, and testing procedure considerations are combined to form a "Test Matrix" rank-ordering the possible choices of horsepower absorbers for a given test. A possible configuration for test facilities utilizing all types of devices is given along with a discussion of possible data incompatibility issues.

TABLE OF CONTENTS

CHAPTER I: INTRODUCTION TO THE PROBLEM 1

CHAPTER II: HISTORICAL PERSPECTIVE ON THE HORSEPOWER ABSORBER 4

CHAPTER III: HORSEPOWER ABSORPTION DEVICES 7

Hydraulic Dynamometer 7

Electric Dynamometers 14

Air Dynamometers 20

CHAPTER IV: TURBOSHAFT ENGINE TESTING 26

Test Standardization 27

CHAPTER V: TURBOSHAFT ENGINE APPLICATIONS 32

CHAPTER VI: ANALYSIS AND RECOMMENDATIONS 36

CHAPTER VII: CONCLUSION 42

APPENDIX

DIMENSIONAL ANALYSIS 44

LIST OF REFERENCES 48

BIBLIOGRAPHY 50

LIST OF FIGURES

Figure 1 Prony brakes 4

Figure 2 Typical water brake 7

Figure 3 Operating envelope for typical water brake 8

Figure 4 Cavitation damage to water brake exit flange 9

Figure 5 Water brake with multiphase flow inside 11

Figure 6 Closed circuit water brake system 12

Figure 7 Power vs time curves for typical water brakes 13

Figure 8 Midrange loading power vs time curves for
DC generator as dynamometer 15

Figure 9 Zero to maximum load power vs time curve for
DC generator as dynamometer 16

Figure 10 Three DC dynamometers in series with resistance rack 17

Figure 11 Schematic of eddy current dynamometer 18

Figure 12 Two views of eddy current dynamometer 19

Figure 13 Eddy current dynamometer power vs time curves 20

Figure 14 Fan brake, with loading curve 21

Figure 15 Centrifugal compressor, with inlet guide vanes 22

Figure 16 Axial compressor 22

Figure 17 Variable Radial Outflow Compressor (VROC) wheel and shroud 23

Figure 18 VROC dynamometers 24

Figure 19 VROC operating envelope 25

Figure 20 Water atomizers for icing tests 27

Figure 21 Allison T53 - A - 720 mated to an AVCO/Lycoming water brake 30

Figure 22 Power vs Time curve for refused simulated forced landing 32

Figure 23 Power vs Time curve for acceleration for fixed wing take off 33

Figure 24 Power vs Time curve for refused fixed wing take-off 33

Figure 25 Power vs Time curve for armored vehicles 34

Figure 26 Power vs Time curve for watercraft 34

Figure 27 8W NAWCAD Test Cell Gearbox -- showing multiple water brakes 37

Figure 28 GE T700 engine mated to a NREC 2500 BHP air dynamometer 38

Figure 29 Lycoming T53-L-703 turboshaft mated to AVCO-Lycoming water brake
in a Modular Engine Test Stand (METS) at Ft Campbell KY. 39

NOMENCLATURE

Symbol	Definition	Units
BHP	brake horsepower	
BTU	British Thermal Units	
EMF	Electromotive Force	
HP	horsepower	
NPSH	net positive suction head	(ft)
NREC	Northern Research Engineering of Cambridge	
PFRT	Preliminary Flight Rating Test	
RLC	Rapid Load Change	
VROC	Variable Radial Outflow Compressor	
D	rotor diameter	(ft)
H	head	(ft)
Q	flow rate	(gal/min)
T	Temperature	(Rankine or Fahrenheit)
q	dynamic pressure	(lbf/sqft)
r	rotor radius	(ft)
P_a	static pressure	(lbf/sqft)
P_t	total pressure	(lbf/sqft)
P_v	the vapor pressure of the fluid medium	(lbf/sqft)
ε	surface roughness	(ft)
ρ	fluid density	(lbm/cuft)
ν	kinematic viscosity	(lbm-s/sqft)
ω	rotational velocity of rotating component	(rad/s)
σ	Thoma's constant	

CHAPTER I

INTRODUCTION TO THE PROBLEM

After a forty year hiatus, the US Air Force's Arnold Engineering Development Center (AEDC) in Tullahoma Tennessee, is planning to begin turboshaft engine testing during FY 1998. Because of the US Government's Base Realignment And Closure (BRAC) structuring, the Naval Air Warfare Center, Aircraft Division's (NAWCAD) facility in Trenton, New Jersey is shutting down, and transporting the majority of it's turboshaft test equipment to AEDC. Realizing that they had little turboshaft experience, the officials at AEDC commissioned a search for information on turboshaft engine testing. The search started using turbojet testing as a guide, since AEDC has considerable experience in that area, and has been a leader in standardizing the field. However, it soon became apparent that test requirements for turbojet and turboshaft engines were vastly different. It was found that, as opposed to thrust measurement in turbojet testing, the most critical requirement for turboshaft engine testing is reliable, precise, and controllable horsepower measurement and/or absorption. Establishing a satisfactory turboshaft engine test requires understanding the characteristics of the horsepower absorber in a given test, and how those characteristics change for different test conditions and engine applications. After considering the scope of the problem, and conducting an exhaustive literature search, it was determined that no single document existed that specified a particular turboshaft engine application and test condition for a particular horsepower absorption technique from both a qualitative and quantitative perspective. Therefore, this thesis was initiated to provide such a document. There have been several proprietary attempts to address this problem, mostly by engine manufacturers desiring an economical method of engine testing. Most of these tend to focus on adapting new devices to existing procedures. In the academic world, the only source found that described the same sort of investigation was written by N. C. Baines from the Imperial College in London in 1992 (see Bibliography).

The organization of this thesis is as follows: Chapter Two provides a historical perspective of horsepower absorbing devices. The problem of accurately measuring horsepower output and safely

absorbing that horsepower has been present since the beginning of the industrial revolution. Early techniques are introduced and discussed.

Chapter Three analyzes the current horsepower absorbers in use. Hydraulic Dynamometers, Electric Dynamometers, and Air Dynamometers are discussed in each of their currently useful forms. Each device's operating principles are reviewed, and both positive and negative design and operating characteristics are discussed so that trade-off factors become readily apparent. As one might expect, each type of dynamometer has an operating region where it is the optimal device to use; therefore a test engineer need use only one type device in a narrow operating band to perform a specific test. There are also regions where any of the dynamometers will suffice; only cost or previous experience can be used to discriminate between devices.

Chapter Four discusses Military Specification (MIL-E) -8593A, "Engines, Aircraft, Turboshaft and Turboprop, General Specifications For," dated 15 October 1975 and several other testing sources. The differences are delineated between Development, Qualification, and Acceptance testing. Qualification was chosen as the representative test because it encompasses the entire spectrum of engine testing. A Qualification test is summarized from of MIL-E-8593A and used to analyze each of the horsepower absorbing devices for their applicability to given types of tests.

Chapter Five qualitatively examines turboshaft engine applications. This chapter represents the crux of the problem. The single most important piece of information in deciding a particular horsepower absorber over another is the engine's final application. Different machines use engines differently. Helicopter engine loading characteristics are dual transient, while fixed wing engine loading characteristics are single transient. Automotive applications are concerned with specific fuel consumption, while watercraft require another sort of dual transient exactly opposite that of helicopters. Thus, some of the horsepower absorbing devices are better for modeling certain load and inertia characteristics than others. As can be expected, none are applicable to all test conditions.

Chapter Six combines all the above considerations into a single "Test Matrix" in an effort to provide an ideal solution to the turboshaft testing activity. Applicability, cost, maintenance, complexity,

repeatability, and compatibility are discussed in an effort to articulate the needs of the real world, and to provide a perspective on decision and selection criteria in cases where no single device is pre-eminent.

Chapter Seven presents a brief conclusion, and five questions to answer when selecting the proper horsepower absorber for a given test.

CHAPTER II

HISTORICAL PERSPECTIVE ON THE HORSEPOWER ABSORBER

The term "horsepower absorber" is actually a misnomer. These devices really do not "absorb" horsepower, they convert it to thermal energy through various force interactions such as friction, viscous shear, momentum transfer, or electro-mechanical force interaction. Sensors exterior to the device such as force balances, torque deflection, and rotational speed measurement devices quantify the horsepower dissipation. In this investigation the term "horsepower" is used because of the device convention, not because of any unit preference. One can easily substitute "power" for "horsepower" and not lose any context.

Until the late nineteenth century, a simple friction device called a Prony Brake, see Figure 1, was used to provide a mechanical friction to oppose rotation while testing power sources. In the left figure, the

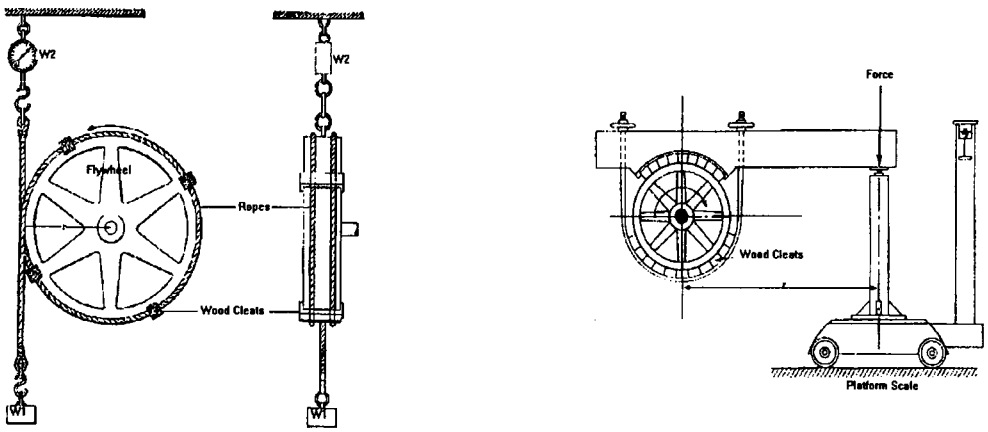


Figure 1 Prony brakes
From: Beckwith, Thomas, et al. Mechanical Measurements. Reading MA: Addison Wesley, 1968 p 436

device is a wheel on an axle common to the engine, around which a rope is wrapped. Wooden cleats constrain the rope to the wheel. When the engine turns the wheel there is friction between the rope and wheel that tends to oppose movement. The downward force generated on the rope registers on the spring scale above the device. That force subtracted from the weight on the opposite end of the rope yields the

net force, multiplied by the radius of the flywheel gives torque, multiplied by rotational or angular frequency gives power. A similar apparatus is the friction brake in the right figure. These devices absorb horsepower according to a simple equation:

$$\text{Brake Horsepower} = \frac{2\pi rnw}{33,000} \quad (1)$$

Where: r = the length of the brake arm, if feet

n = revolution of the shaft per minute (rpm)

w = weight of the brake arm in pounds

The 33,000 is foot-pounds per minute per horsepower (Beckwith, p 147).

These two devices are representative samples of mechanical friction brakes used during the early years of the Industrial Revolution. The operating envelope for these devices is typically what one would expect for an early steam engine -- several hundred horsepower at a maximum of around 1500 RPM. Their advantages are that they are simple, inexpensive, relatively easy to maintain, contain few moving parts, do not require specialized materials or support facilities/equipment, and do not require highly skilled labor to operate or maintain. Their disadvantages are that they are fragile, frequently flying apart before a given test was completed, and that they were made from combustionable materials. Given that the energy is dissipated as heat, these devices could catch fire easily if not properly cooled; thereby presenting a serious safety hazard. Judging from the available literature, the devices usually did not last more than one test.

In the late 19th and early 20th Century, there was an explosion of literature on the subject of horsepower absorption. Most of the physical principles and basic techniques used have not changed in almost 100 years; this is not really a new science. The principle is simple: how to turn mechanical energy into thermal energy, and carry the thermal energy away from the device before the device is destroyed. Changing mechanical energy into thermal energy can be accomplished several ways. Two of these are mechanical or fluid friction and electromotive force (EMF). There are many devices that can do this, some more efficiently than others. To transfer this thermal energy from the device to the surroundings, some

thermal energy absorbing material is required. This material must have some essential properties to be effective: it must be easily exchanged without effecting the device's continuous operation, it must not change its fundamental absorption properties once it has been heated and cooled so it can be reused, it must have a specific heat compatible with conventional handling and cooling systems, it must not require any elaborate storage or transport considerations, and it must be plentiful and inexpensive. Solid substances can be ruled out as, given the Prony Brake examples, they are too difficult to exchange during the normal operation of the device, and frequently do not maintain their fundamental properties after heat exchange. Some sort of fluid then, appears to be what is required. When working in British Units, and using fluid as a thermal energy absorbing material, a rule of thumb for estimating power absorbed (in HP) is:

$$\dot{W} = C_p \dot{m} \Delta T \qquad (2)$$

In units of

BTU/min = (fluid sp ht) x (flow rate in lb/min) x (fluid ΔT through device in ^{o}F)

Knowing that 100 HP = 4,244 Btu/min yields power (in HP) as a function of temperature difference. A similar relationship is available in SI units (Bronder p 8).

CHAPTER III

HORSEPOWER ABSORPTION DEVICES

Hydraulic Dynamometer

While possible fluids for use as a thermal energy dissipation medium range from air to oil, one must consider the venue of the engineers first trying to solve this problem. In the last part of the 19th Century, the technology to use exotic fluids was simply too complex to be economically viable. Air was used in some low horsepower applications by some windmill-like devices, but the material technology to exploit it was simply too far in the future. The engineers of the era solved the problem by using the most practical fluid commonly available -- water. Even today, the most common technology used for horsepower absorption is the hydraulic dynamometer, referred to as a water brake, pictured in Figure 2.

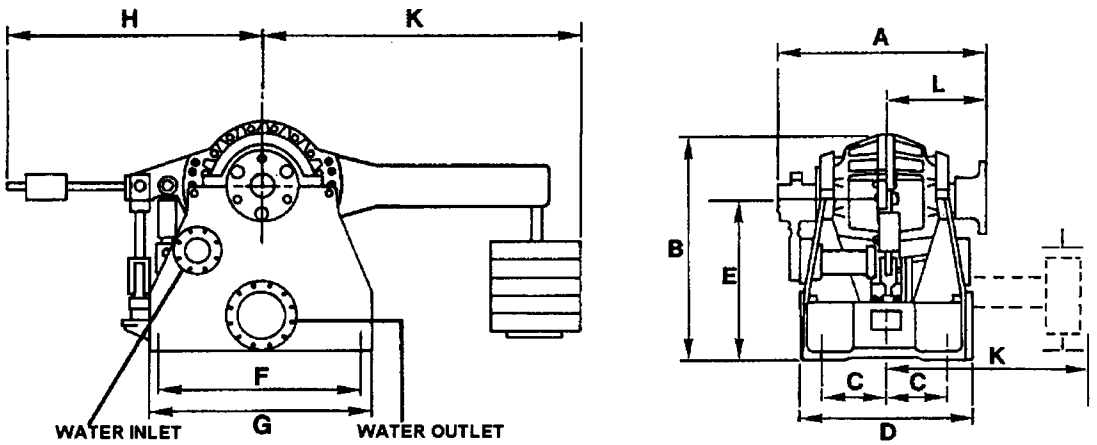


Figure 2 Typical water brake

From: F Range Dynamometers. Worcester, England: Froude Consign Industries LTD, 1990.

Invented by Froude in the 1870's, water brakes have changed little from their inception. Typically, water brakes consist of a cylindrical chamber containing one or more perforated or cupped discs or rotors on an axial shaft which rotate between sets of perforated or cupped stators. In most applications, water enters the chamber axially and is accelerated by the rotating components and thrown outwards by centrifugal action. A favorable axial pressure gradient is maintained within the device to ensure axial flow. From the outer diameter of the chamber inwards, the water forms an annulus which rotates at approximately half the rotor

speed. The momentum imparted to the water annulus by the rotor is absorbed by the stator. The energy dissipation is due to viscous shear forces, both laminar and turbulent, acting on the physical surfaces of the rotor and stator, and the mean and turbulent motions of the fluid eddies dissipating the kinetic energy of the fluid motion. Momentum transfer between the rotor and stator is also generated by viscous shear forces and by pressure fields generated by the rotor and stators inside the device housing. From this, there results the torque, and torque transients, on the rotors that the input horsepower overcomes, and causes a reaction force on the stators that tends to turn the housing. The horsepower absorbed (load applied) is thereby a function of the wetted area within the device. The housing is restrained from rotating by a load cell which is mounted to a torque arm a known fixed distance from the device centerline, thereby allowing torque measurement. The waste heat is, of course, carried away by the water flow.

Water brakes come in many different configurations and ratings and are used to test everything from aircraft engines to emergency electrical generators for hospitals. The operating envelopes for the family of devices pictured in Figure 2 is depicted in Figure 3.

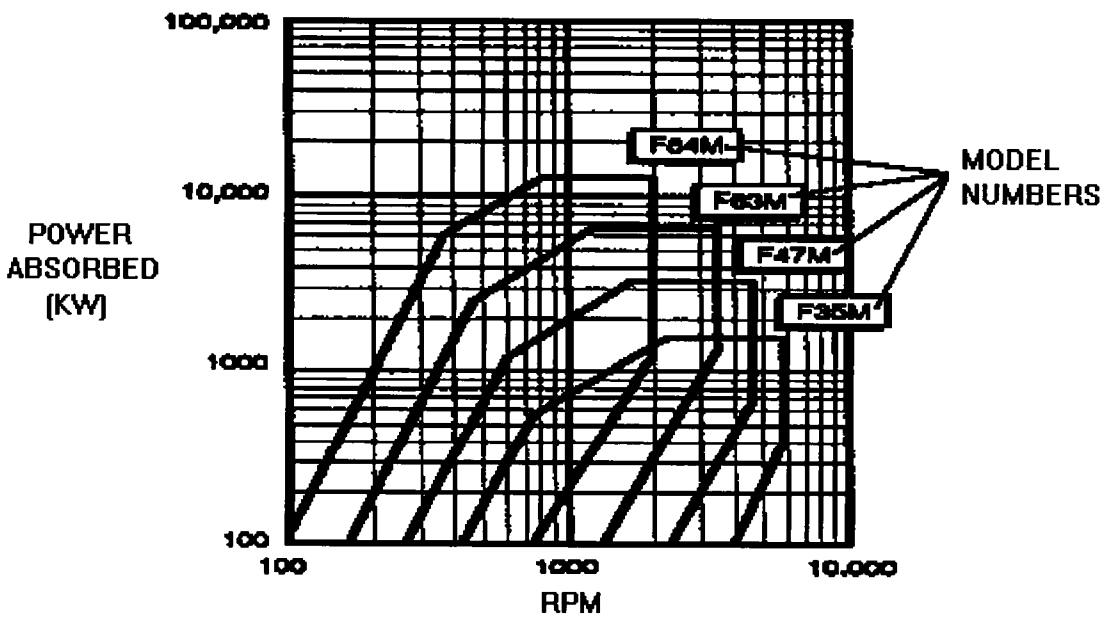


Figure 3 Operating envelope for typical water brakes

From: F Range Dynamometers. Worcester, England: Froude Consign Industries LTD, 1990.

Despite the many varied uses, sizes and manufacturers of water brakes, a common problem that is frequently overlooked as a design criteria is cavitation. When the static pressure at certain locations inside the dynamometer housing drops to a point equal to or below the fluid vapor pressure, the fluid at that location will vaporize. The formation of these vapor bubbles is, by itself, not harmful. It is the collapse of these bubbles on to a surface inside the device that can often cause catastrophic damage. Figure 4 shows some typical cavitation damage to a water brake exit flange.

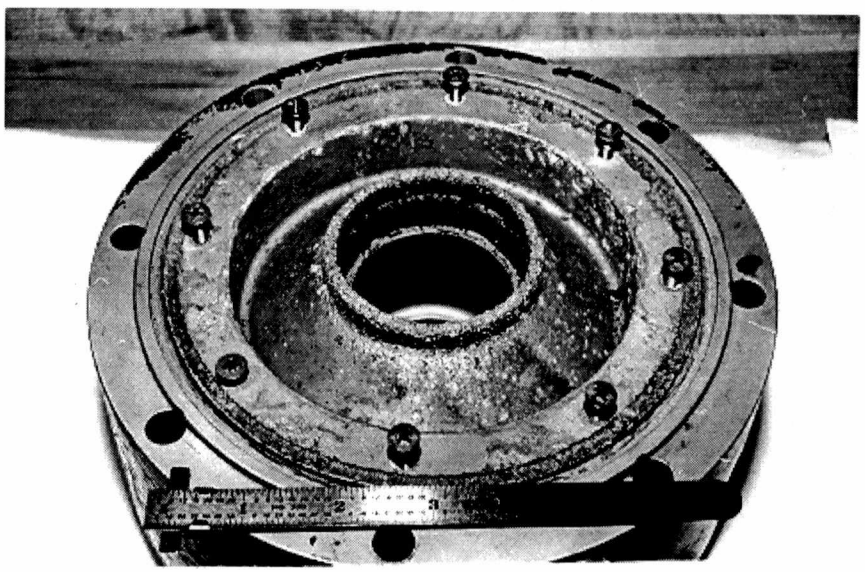


Figure 4 Cavitation damage to water brake exit flange

Most water brakes are designed for horsepower absorption at a specific flow rate and operating rpm, then the operating pressures are adjusted to work around excessive cavitation. Every device can cavitate to some extent. If the level of cavitation is high, catastrophic failure can result in a matter of minutes. At a minimum, cavitation severely erodes the critical parts of the device, and can drastically shorten the effective lifespan of internal components if not closely controlled. Cavitation is a form of multi-phase flow which frequently occurs in or around other types of turbomachinery such as boat propellers and rotary pumps. Most turbomachinery modeling uses Thoma's Law of Similarity and Head

Coefficient for geometric similarity, and Power or Flow Coefficient for dynamic similarity to develop prototype dimensions (see dimensional analysis in appendix). Water brakes are no exception. However, Thoma's Law is not sufficiently accurate to insure cavitation free operation. Theoretically, the device will cavitate at the location of the lowest static pressure. This occurs at the backside of any protuberance into the flow due to flow separation -- the leading edges of holes in the rotor disc, the trailing edge of a rotor tip, or the edge of a rotor disc. The challenge is to maintain the working cavitation number at a value greater than some critical cavitation value.

Analysis

Using British Units, flow rate required through the device is established by an allowable temperature rise of the working fluid, governed by the equation (Kahn, p 32):

$$Q = \frac{(5.09)\mathrm{HP}}{\Delta T} \tag{3}$$

where temperature is measured in degrees Fahrenheit. A rule of thumb for this, however, when using water is 60 gpm per 1000 hp of dissipated load (Bronder, p 8). This flow rate is required to effectively carry away the heat generated by the horsepower conversion and maintain a constant outlet temperature. Most water brake applications utilize a closed-circuit water cooling system that is capable of maintaining outlet water temperature +/- 10° F.

In most applications, it is the difference between the static pressure and the vapor pressure (P_s - P_v) that is responsible for cavitation (see cavitation number in appendix). Given a particular test condition, water condition, horsepower absorption, and device configuration, ρ and ωr are usually constant, however, (P_s - P_v) is usually erroneously assumed as nearly static throughout the operating range. In a given fluid, vapor pressure is a function of temperature -- fluid temperature goes up, vapor pressure goes up. In a water brake, this means the higher the horsepower absorption (loading) the higher the ΔT across the device, the higher the vapor pressure, and the more cavitation present. Water condition can also effect vapor pressure. If there are dissolved hydrocarbons, gases (like air) or minerals within the water, the vapor pressure rises

even more. These factors cannot be accounted for, in general, in testing or operating power absorbers. This is the reason that the cooling systems and water condition are so critical in water brake operation.

The control instability characteristic of water brakes is caused by the fluctuating boundary between the liquid and vapor water within the device, which changes the wetted area. The reason for this multiphase flow is the pressure differential created by the water annulus -- the lower static pressure is around the central shaft causing the fluid to vaporize. Given that the vapor pressure, P_v, can vary significantly over a test condition due to the temperature gradient across the device --especially if the water is not cooled properly, the only way to control (P_s - P_v)and hence cavitation, is to raise the static pressure, P_s, accordingly. As a water brake's load characteristics are a function of its wetted area, unpredictable multiphase flow means an unstable load characteristic. The problem is how to control P_s, and be able to correlate a change in P_s to a decrease in cavitation potential. It is usually not possible to visualize the flow inside the device due to the modifications necessary (Courtney p 1). However, were one able to do so, the annular flow pattern pictured in Figure 5 would appear. It is because of this multiphase flow that many

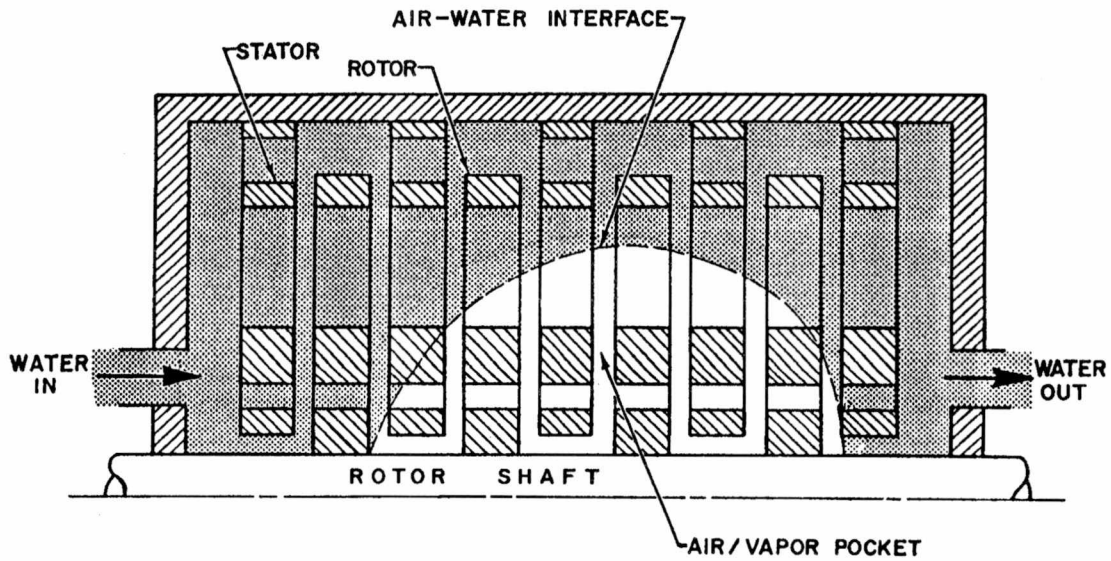


Figure 5 Water brake with multiphase flow
From: Courtney, W. J. Analysis, Test and Design of a Water Dynamometer. Philadelphia PA: Naval Air Engineering Center, 1969. p53

water brakes employ some internal pressure regulating mechanism as a form of cavitation control (usually a pressure port on the housing connected to shop air). Increasing the static pressure inside the housing will shrink the multiphase flow region, and decrease overall cavitation. This technique is limited by the housing operating pressure, however. Because equation (2) illustrates that temperature increase across the device is a function of horsepower absorbed, given their higher ΔT, water brakes designed to absorb higher horsepower would encounter a maximum internal pressure limit while trying to control cavitation due to an increase in operating temperature. Clearly, if higher horsepower shaft engines are to be run, then either a very high rate heat exchanger, or a continuous supply of cool water, is required. A typical cooling system for a water brake is pictured in Figure 6.

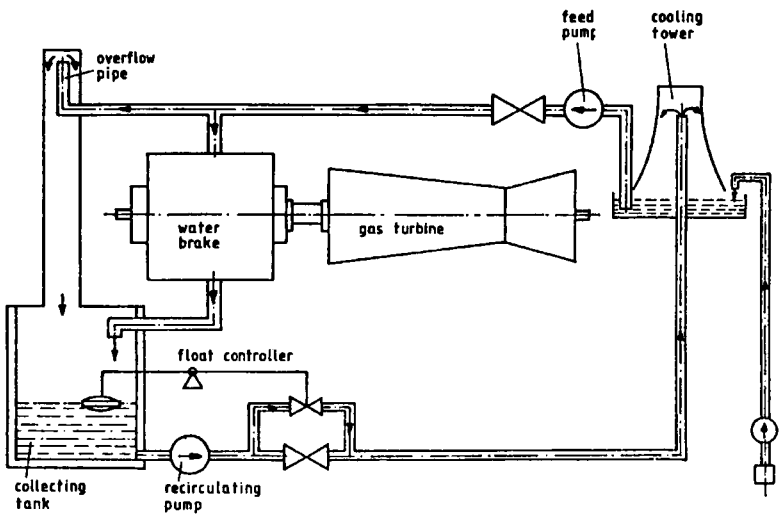


Figure 6 Closed Circuit Water Brake System
From: Bals, Herbert, et al. Full Load Testing of Gas Turbines up to 120MW. Ruhr, Germany: Krafterwerk Union, AG, 1978, p 14.

The irregularities introduced by the many sources of error in this system (dissolved air and hydrocarbons in the water, inaccurate temperature measurement, nonuniform surface roughness etc.) can also compound the problem -- usually by decreasing the cavitation number at a given operating point or by changing the systems load characteristics.

Water brakes are controlled by a series of valves in their fluid supply piping. An inlet valve and an outlet valve are normally utilized. Operating points and loading procedures are dictated by the device's

load characteristic, but there are some general trends: When operating a device, the minimum horsepower absorption setting corresponds to a minimum amount of water in the housing necessary to lubricate the bearings with the inlet opened only enough to ensure flow, and the outlet valve full open. To increase the load, the inlet valve is progressively opened, and the housing slowly filled. Once the housing is filled, and the inlet valve full open, the outlet valve may be further restricted to increase the pressure and hence the load -- sort of like a pump having to work harder against an upstream pressure gradient. It is this flow restriction at high load that causes the trade-off in flow temperature and device pressure. When considering small time intervals (2-5 seconds) water brakes are stable on the "power up" side of their operating curve, easily loading from a minimum tare load of approximately 5% of maximum to maximum loading. Instability, however, can arise when, again considering small time intervals, we operate on the "power down" side of operating curve. Ideal power vs time curves are pictured in Figure 7. To power down

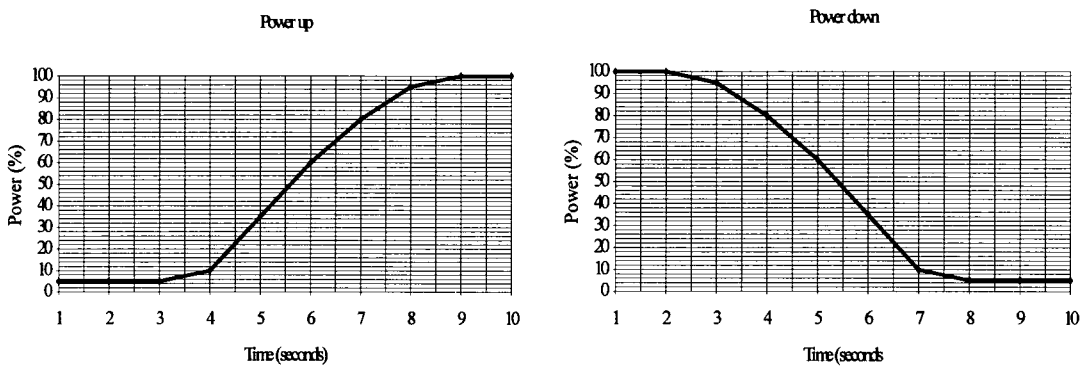


Figure 7 Power vs time curves for typical water brakes

there are two options on valve positioning. First is to reverse the process of the power up, opening up the outlet valve until fully open, then closing the inlet valve until reaching the tare load. The second is to begin closing the inlet valve until a suitable pressure exists in the device, then opening the outlet valve as appropriate. Performed quickly, the first method risks losing static pressure in the device too quickly, thereby inducing cavitation, and forming the central vapor core illustrated in Figure 5. The second method risks raising the temperature of the water due to decreased flow, again risking cavitation and multiphase flow due to increased vapor pressure. When the device cavitates uncontrollably, the central vapor core

exposes too much of the rotors, decreasing the devices wetted area unpredictably, thereby causing the device to become unstable.

It is important to note that, as illustrated in Figures 3 and 7, because water brakes have a tare load requirement related to a minimum water level, higher rpm, lower loading conditions are outside the device's envelope, as are rapid unloads from a maximum condition. There are complex control systems that can be used to "engineer around" these defects, most notably by using a high speed valve switching mechanism controlled by a computer sensing temperature and pressure changes within the housing. Apparent advantages to using water brakes are that they are plentiful, inexpensive, do not require highly skilled labor to maintain, and there is a wealth of empirical operating knowledge available. Disadvantages are that they become unstable when operated outside of their envelope (they are very unforgiving of even the slightest deviation), require a sophisticated control mechanism to accurately model a time varying load, require a large water support system (cooling, pumps, hoses, filters, antifreeze, etc), and are very susceptible to cavitation damage and subsequent catastrophic failure.

Electric Dynamometers

Electric Dynamometers have been used in engine testing since the 1930's. Almost any rotating electric machine can be used as a method of horsepower absorption. There are four machine types feasible for use as dynamometers: DC generators, DC dynamometers, AC motors and generators, and Eddy Current dynamometers. It should be cautioned, however, that while almost any general purpose rotating electrical machinery can be adapted for use as a dynamometer, the results are usually inferior to equipment designed specifically for use as a dynamometer.

DC Generators

Ordinary DC electric generators may be used as dynamometers. By rigidly mounting the device, and driving it with the test engine causing the armature to rotate in the magnetic field formed by a set of

permanent magnets, DC electric current will be produced. The DC electric current will be proportional to the rotational speed. The power absorbed (in watts) can be computed using this equation:

$$\text{Power} = \frac{EI}{\eta} \tag{4}$$

Where Power = the power (in watts) absorbed

E = output voltage, in volts

I = output current, in amps

η = the efficiency of the generator (Beckwith, p 440)

The device is loaded by applying some electrical load, possibly a graduated resistance rack. Using this method, the devices are very easy to control. Both "power up" and "power down" characteristics are similar, and they have the ability to operate in the high RPM, no/low load region. They are relatively easy to maintain, safe, reliable, plentiful, and varied in design. They also dissipate the thermal energy produced external to the device. This technology, while seemingly elegant, can be fraught with modeling and measuring difficulty, however. DC generators tend to be variable effective inertia machines. The inertia they appear to apply to a test engine may differ based on the amount of load already applied, and the RPM of the device at the time of loading. The efficiency of DC generators also varies the same way. For instance, if we want to increase from a mid-range load condition, to a high load condition, we could simply add a bank of resistors, and the power vs time characteristic could look like Figure 8

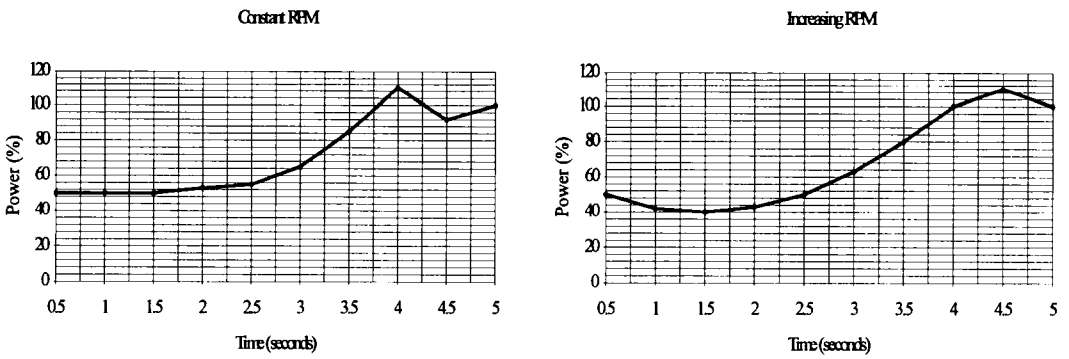


Figure 8 Midrange loading power vs time curves for DC generator as dynamometer

However, if we started from a no-load condition, the power vs time characteristic could look like Figure 9.

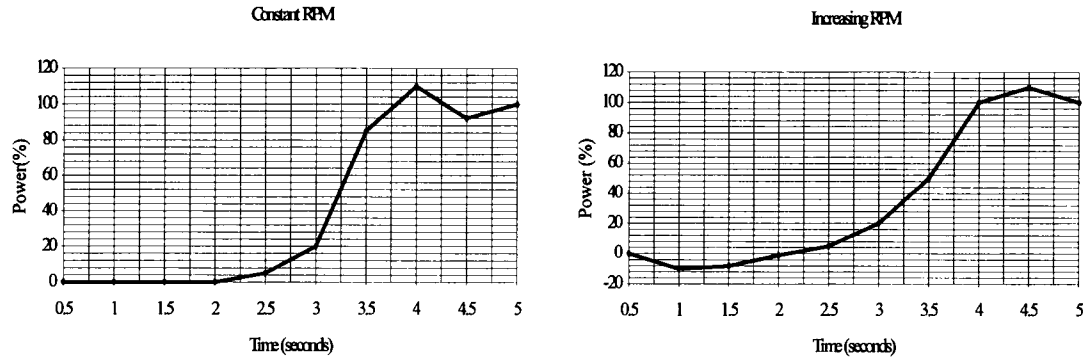


Figure 9 Zero to maximum load power vs time curve for DC generator as dynamometer

Note that the curve slopes are different. Inertia tends to increase, and efficiency to decrease, as the device loads at constant RPM. If we were to increase RPM as we increase load, effective inertia would tend to decrease and efficiency to increase. The device also does not accurately model a short time interval, rapid load change (RLC). In figures 8 and 9, there is a marked lag after initial loading (the apparent driven region on the increasing RPM data is most likely due to data scatter). This lag is common to most engine/dynamometer systems. However, the lag portion of the power vs time curve tends to be too shallow and broad to accurately model turboshaft engine transient response and frequently sets up a sympathetic resonance with engines equipped with hydro-mechanical fuel control units. The load curve will shift out of phase from the engine power curve, causing each device to "chase" the other. Unless the inertial and efficiency characteristics are known exactly, these devices make accurate load modeling difficult.

DC Dynamometers

DC Dynamometers are simply DC generators designed with more predictable inertial and efficiency characteristics. Because of these design considerations, they usually make poor generators. DC dynamometers have many of the positive attributes of water brakes and DC generators, but without the instability problems of water brakes, and the differing inertia characteristic of DC generators. As with the water brake, DC Dynamometers are also cradled in trunion bearings, permitting direct reaction

measurement. As with DC generators, the DC current generated by the load absorption is dissipated as thermal energy in graduated resistance racks located external to the device. Since thermal energy is usually destructive to most dynamometers, the value of this feature cannot be overstated. Figure 10 shows some typical DC Dynamometers in series with resistance rack.

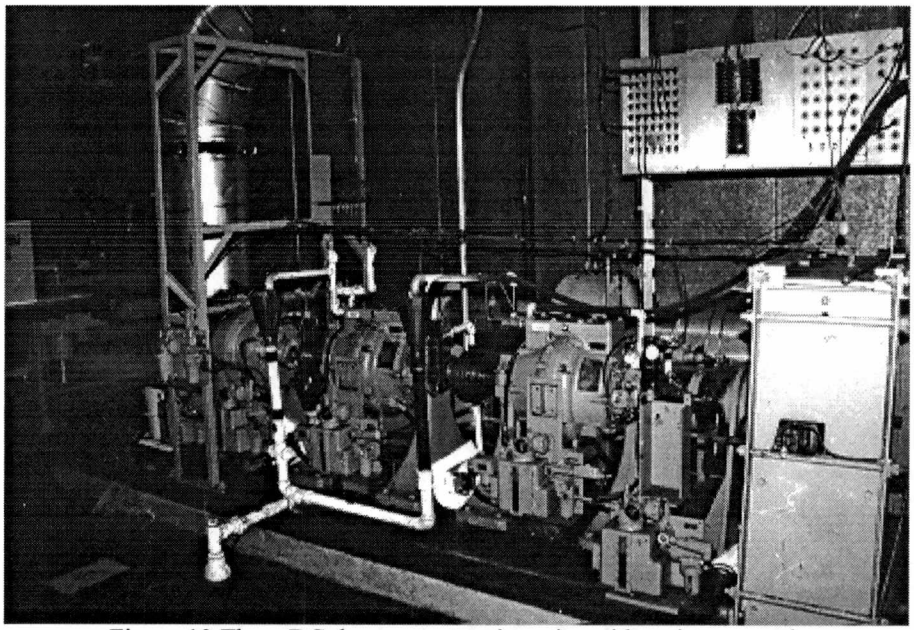


Figure 10 Three DC dynamometers in series with resistance rack

The devices are simple to control, and offer predictable performance over their operating envelope, including the high RPM, no/low load condition. However, as with the DC generator, transient loading characteristics are sluggish. While exhibiting a more predictable inertia characteristic, the device does not track an RLC accurately, and can set up the same lagging phase resonance with hydro-mechanical fuel control equipped engines. Currently there is only one manufacturer of DC Dynamometers, who makes only one type of device capable of absorbing loads in the 4000 bhp range. Because of the lack of extensive empirical knowledge, DC dynamometers have historically been used only in limited applications.

AC Motors and Generators

AC motors and generators, while commonplace and feasible for use as dynamometers, require considerable control and rectifier circuitry. Even when closely controlled, their capabilities are often

inferior to DC systems with respect to slower response times, mechanical failure, and load characteristics. Their use would be justified only when other devices are not available.

Eddy Current Dynamometers

An Eddy Current Dynamometer is a different sort of machine. In a DC generator, a wrapped wire conductor forming a complete circuit moves through a magnetic field, using a commutator, and current flows through the circuit as a result. An Eddy Current Dynamometer on the other hand, does not use a complete circuit. A drum is substituted for the wire, and is not allowed to complete a circuit. The drum, usually made of a ferrous material, is attached to the engine driveshaft and rotates in the field of stationary electromagnets. The relative motion between the drum and the electromagnets induces the eddy currents (essentially short circuits) in the drum material. The eddy currents are unable to flow through a complete circuit external to the device, and are confined within the drum. Confining these currents creates an electromagnetic field within the drum which tends to oppose the applied field, thereby generating the braking force. These currents are dissipated as thermal energy within the device, which is carried away by a cooling fluid, usually treated water. The device load is controlled by the amount of current applied to the electromagnets. More current means a stronger applied field, which produces stronger eddy currents, thereby producing a stronger induced field, increasing the braking force. Figures 11 and 12 illustrate typical eddy current dynamometers.

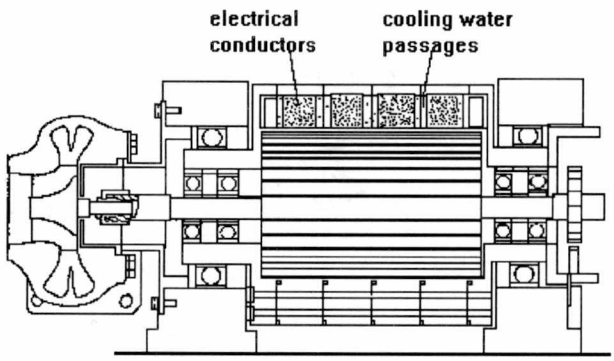


Figure 11 Schematic of eddy current dynamometer
From: Baines, N.C. "Testing Techniques." In Radial Turbines, Lecture Series 1992-05. Rhode Saint Genese, Belgium: Von Karman Institute for Fluid Dynamics, 1992, p 15.

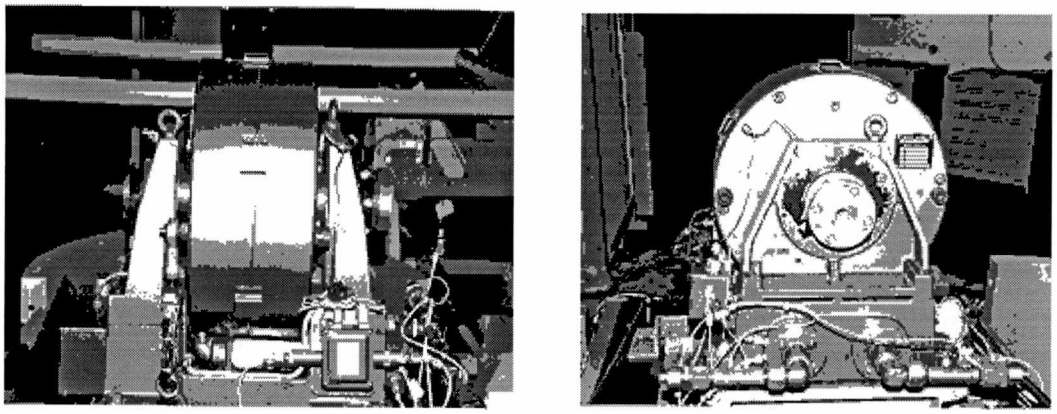


Figure 12 Two views of eddy current dynamometer

The drum requires a material with a high magnetic permeability, high resistivity and a high strength at the working temperature As one would expect, the operating envelope of these devices is limited by the amount of heat that the drum can dissipate; therefore cooling both the drum and the electromagnets is a significant design challenge. The cooling jacket frequently is placed between the electromagnets, surrounding a circular housing containing the rotating drum. There are two primary types of these devices: air gap and wet gap. The air gap type has air between the drum and the housing, relying on convection and radiation to cool the drum. The air cools the drum, the cooling jacket cools the air, and absorbs the thermal energy through radiation. The wet gap design circulates liquid cooling fluid between the drum and the housing, relying on convection and conduction in the cooling fluid as the primary heat transfer mechanisms.

Because of the few moving parts and simple design, they are highly reliable and demonstrate a very long useful life; over three times that of water brakes. Disadvantages are that, as compared to water brakes, they are heavy, usually 75% heavier for a given operating range. They are more expensive initially, up to 100% more. They not only need the same sort of water treatment and cooling requirements as water brakes, but also require an electrical control system, and a substantial source of electric power to run the electromagnets. They also need to be maintained by skilled workers.

However, the primary advantages of an eddy current dynamometers far outweigh the disadvantages, mainly, they do not have the envelope limitations of water brakes. The high rpm, low load condition is easily simulated and the device is stable on rapid unloading. Other advantages are that the control mechanism is simple as compared to the water brake. The device is easily controlled electrically and therefore can be controlled automatically, usually by a microcomputer. They also have a better transient response characteristic, as illustrated by the differences in between figure 13 and figures 8 and 9.

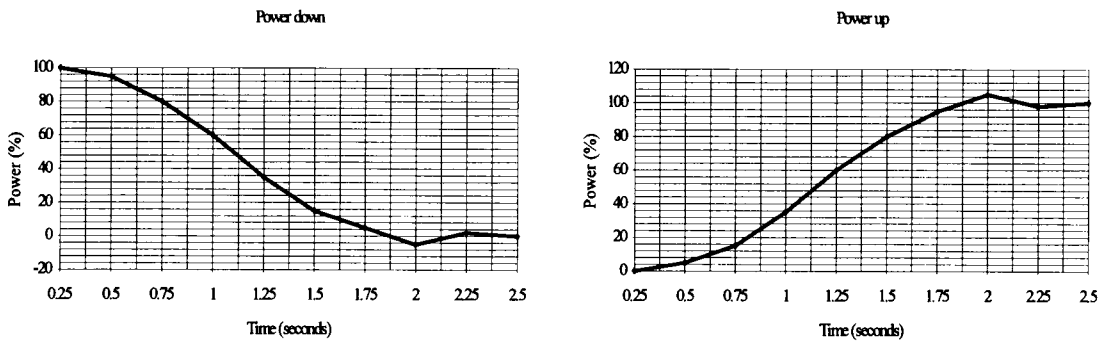


Figure 13 Eddy current dynamometer power vs time curves

Air Dynamometers

Air Dynamometers are the newest addition to the family of horsepower absorbers. Their premise is that by using air as a working fluid, one can eliminate the need for a complicated water or electrical support system, thereby increasing reliability, versatility, mobility, and decreasing required maintenance. The basic principle is simple: dissipate horsepower by moving and subsequently heating a large quantity of air. The more air heated, the more horsepower dissipated (this is the same premise as with the water brake). In the 1970's the US Navy commissioned Northern Research and Engineering of Cambridge Massachusetts (NREC) to conduct an analytical investigation of air dynamometer concepts for future turboshaft engines. In the course of their research, NREC engineers found that several attempts were made in the early 20th century to use air dynamometers, most notably by aircraft engine manufacturers seeking a

light weight powerplant capable of moving large amounts of air. These early aeronautical engineers had constructed a machine known as a Fan Brake, pictured in figure 14.

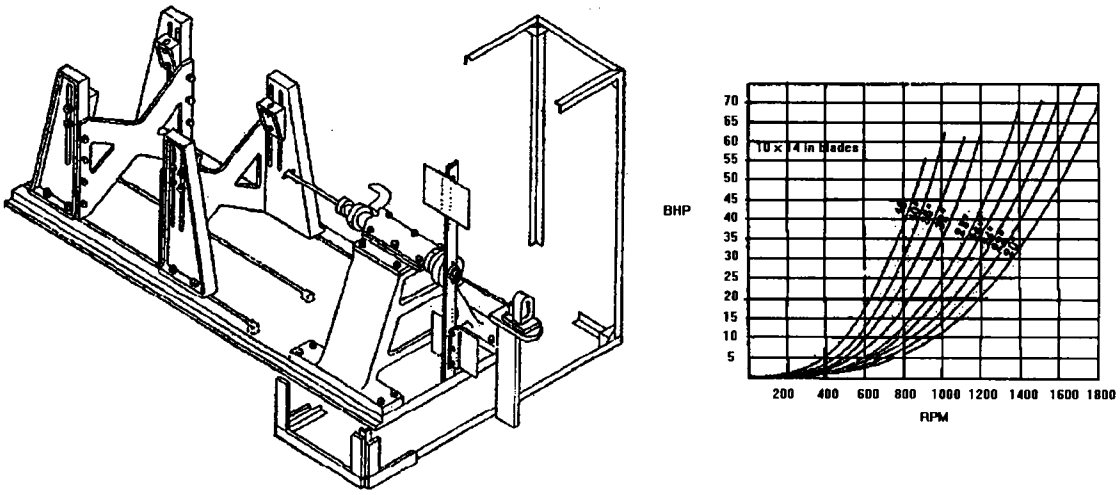


Figure 14 Fan brake, with loading curve

From: Beckwith, Thomas, et al. Mechanical Measurements. Reading MA: Addison Wesley, 1968 p 155

This device used two rotating paddles at the ends of two arms projecting out from a central shaft to dissipate horsepower through aerodynamic drag on the paddles. Using flat plate theory, it is easy to see that the device load is a function of the cube of rotational velocity. By varying the distance between the central shaft and the paddle mounting point, one can vary the device load. Major drawbacks to this device were obvious; the load varied with engine RPM and, seeing as it is prohibitively difficult to move the paddles during operation, could not be varied at constant RPM. The device was also limited in its envelope, as evidenced in the figure. Clearly a device of this nature is not practical in today's venue.

NREC engineers then assumed a compressor to be the primary mechanical to thermal energy conversion device, and researched three categories: dynamic rotary compressors, regenerative compressors, and positive-displacement rotary compressors. Preliminary analysis indicated that only dynamic rotary compressors (centrifugal, axial, and radial-outflow compressors) would be practical because of their efficiency, smooth operating characteristics and controllability (Thirumalaisamy, p 4).

Centrifugal compressors usually consist of a wheel with small vanes that ingest flow radially or axially, drag it along in a circular path, utilizing centrifugal force for compression, then discharging it radially. A turbocharger is an example of a centrifugal compressor. Loading is regulated by varying airflow (mass flow) using a variable orifice, usually controlled by inlet guide vanes. Figure 15 shows a typical centrifugal compressor and associated guide vanes. Centrifugal compressors have as their main

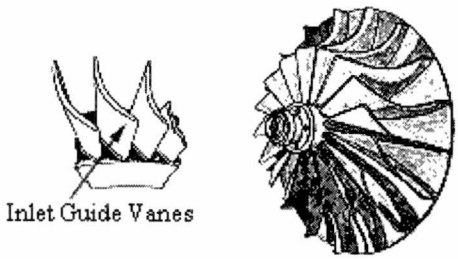


Figure 15 Centrifugal compressor, with inlet guide vanes
From: Driggs, Ivan. Gas Turbines for Aircraft. New York: Ronald Press, 1955, p 119

advantage that they do not surge, even at low speeds. Their disadvantage is that they have a narrow operating envelope. They are also not very cost effective, requiring extensive inlet and outlet controls, thereby increasing system complexity.

Figure 16 shows an axial compressor. Axial compressors consist of a series of wheels, each having a system of radially mounted blades, mounted to a shaft turning within a tapering housing.

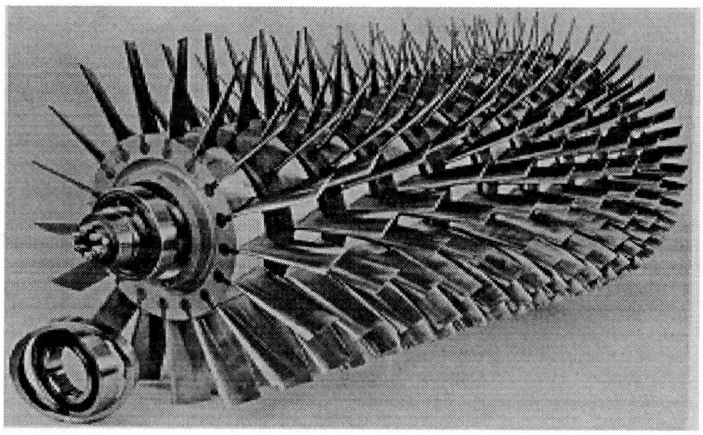


Figure 16 Axial compressor
From: Driggs, Ivan. Gas Turbines for Aircraft. New York: Ronald Press, 1955, p 134

As figure 16 illustrates, an axial compressor looks like a series of fine-bladed fans. Air is compressed by moving through each of the stages, occupying less and less space as it is moved along. Loading is regulated by varying mass flow into the inlet by use of a set of variable inlet guide vanes. The primary advantages of axial compressors are that they are compact, and a significant amount of empirical knowledge exists concerning their design, fabrication and use. Their disadvantages are that they are extremely sensitive to debris ingestion, and surge under RLC.

The device chosen for development was the radial outflow compressor. Radial outflow compressors consist of a shaft mounted wheel with radial compressor blades mounted at the rim of the wheel running parallel to the shaft. The configuration is similar to a paddle wheel, but with blades on both sides of the single wheel. Load is varied by covering, or uncovering the compressor blades, thereby varying mass flow. This action is done with a doughnut shaped shroud that moves axially to the shaft to expose the blades, as illustrated in figure 17. This entire assembly is housed in an axial inlet, radial outflow housing.

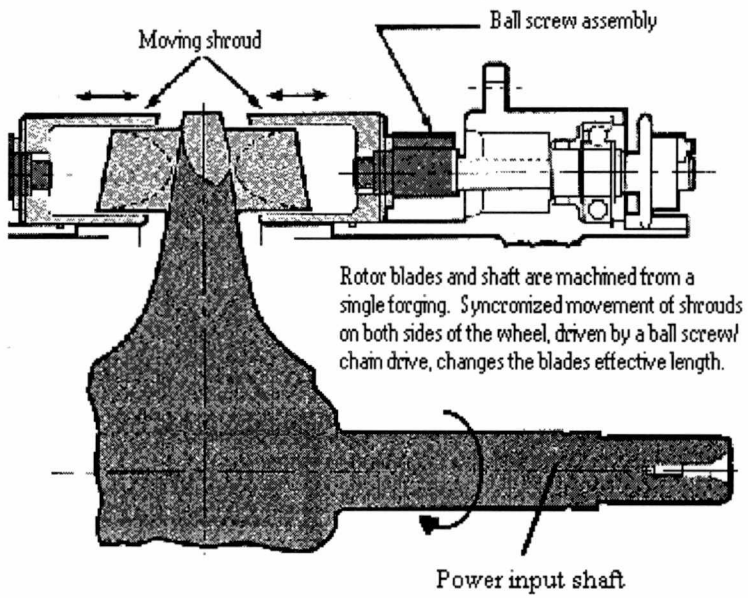
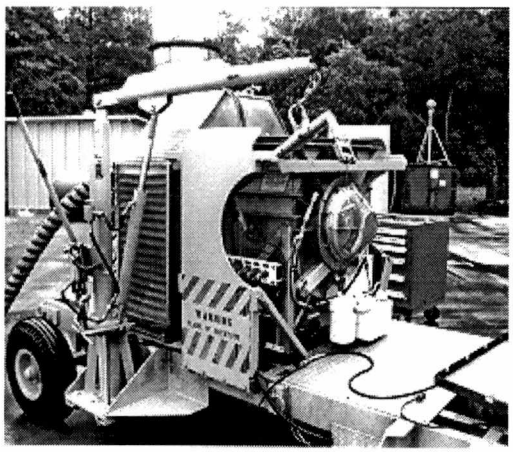
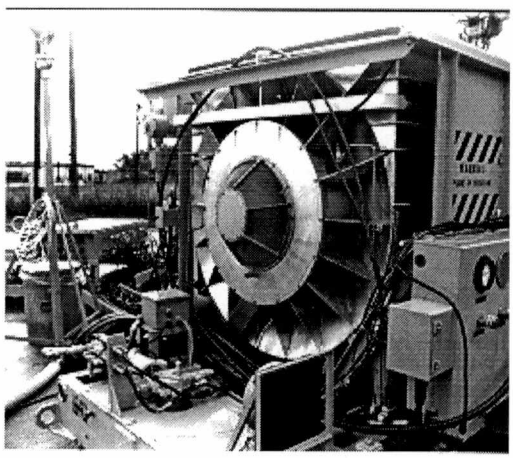


Figure 17 Variable Radial Outflow Compressor (VROC) wheel and shroud
From: Boggs, Robert N., "Movable Shroud Changes Dynamometer Blade's Effective Length." Design News SEP. 1989, p 179

The Navy's objective was to produce a family of devices that could absorb from 500 to 25,000 horsepower at 5000 to 25,000 RPM, yet could be portable enough to be used on a flight line. The NREC study showed that variable radial outflow compressors (VROC) could have a load range of 10 : 1, thereby enabling the use of only 2 devices to cover the entire loading envelope. One device covers from 250 to 2500 horsepower, and the other from 2500 to 25,000 horsepower. Figure 18 shows the two devices.



2500 BHP VROC dynamometer



25,000 BHP VROC dynamometer

Figure 18 VROC dynamometers

Fundamentally, the VROC is a modified version of a centrifugal compressor, and therefore, similar scaling laws apply (Thirumalaisamy, p 12). To absorb the horsepower required in such a relatively small volume requires that a great deal of work be done on the fluid in a short length. To do this, the flow direction must be changed substantially in a short distance. This has been a problem in centrifugal compressors since it leads to low compressor efficiency and instability. In this case, however, efficiency is immaterial, as we want as much loading as possible and do not care about the compression ratio. Stability requirements can be met by regulating the shroud closure rate. The shroud is controlled by a set of calibrated synchronized screws that turn at a controlled rpm. If the device is prone to surging at a particular load setting, then the screw rpm can be varied to compensate for compressibility effects, similar to a bleed band operating on a turbine engine. The main advantages of this device are: It is exceedingly simple -- only requiring four moving parts; the shaft, and three screws that move the shroud. It is also

lightweight, and does not require highly trained personnel to operate or maintain. It is very economical to operate because it does not require large support equipment or consumables (cooling towers, electrical power, pure water etc.). Only two devices are required to cover the entire envelope of turboshaft engine testing, up to 25,000 BHP. It can be operated in a variety of environments: it does not need anti-icing equipment because of its large ΔT; it does not require air cooling or pressurization; it can be operated in open weather, or inside a test cell. Its disadvantages are that it can develop surging if loaded too quickly, for example minimum to maximum load within a second, and it is extremely loud, producing about 170 decibels, due to subsonic/transonic flow in and around the rotor. It does not cover the no load/low load, high rpm condition well as illustrated in the operating envelope in figure 19. It's overall performance falls about mid way between the water brake and eddy current dynamometer.

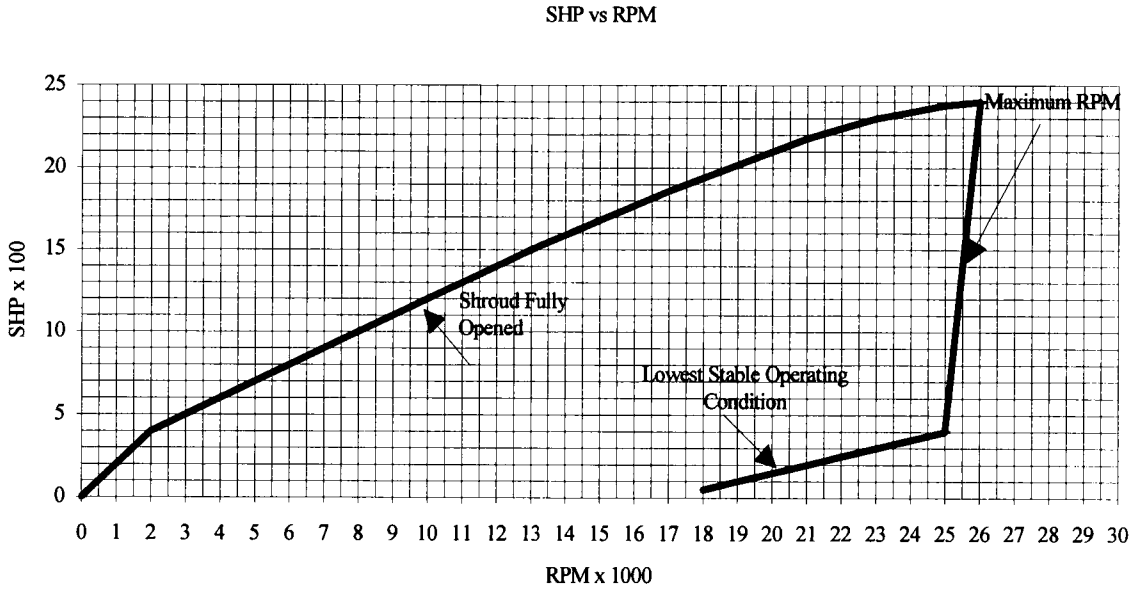


Figure 19 VROC operating envelope
Adapted From: Foor, William B., and William Jansen. The Application of Air Dynamometers for Testing Turboshaft Engines. New York: ASME, 1984, p 7. and Operators Manual, Flexible Engine Diagnostic Stand (FEDS). Washington DC: US Government, 1990, p110.

It also has a large cross sectional area, was designed to be used in the field, not a test center, and must be operated at or near sea level conditions.

CHAPTER IV

TURBOSHAFT ENGINE TESTING

The type of power absorber selected for use is highly dependent on the type of test conditions to be established for the turboshaft engine which includes the type of test conducted, and the environment and load simulated. A device used for steady state testing may, or may not, be suitable for transient testing. The type of turboshaft engine testing done involves four distinct testing categories called: Development, Preliminary Flight Rating Test (PFRT), Qualification, and Acceptance (MIL-E-8593A, p 72)

Development testing is that which is done while a manufacturer is developing an engine for production. It is accomplished using a variable set of parameters. It is usually manufacturer or engine specific, and does not lend itself to general analysis. Product Improvement Testing is a subset of Development Testing which is done when an existing production engine is changed or "improved" to perform a different mission or increase performance. PFRT is the sum of test, demonstration and analysis activity accomplished on engines and components to demonstrate suitability of an engine model for limited use in experimental aircraft flight testing. Qualification Testing is the sum of test, demonstration and analysis activity accomplished on engines and components submitted for qualification to demonstrate the suitability of an engine model for production and service use. Acceptance Tests are those tests conducted on engines submitted for acceptance performance to the extent specified in the engine specification. Performance Variation Testing is a subset of Acceptance Testing which is done to engines just off the assembly line as a quality control check to ensure that the production run is meeting contract specifications. For the purpose of choosing a typical test for analysis and the implications for a horsepower absorber for such a test, a Qualification Test was chosen as the most general.

The test condition is the environment or state within which the particular test engine is operated. Typical test conditions are denoted as Starting/Shutdown, Transient, Acceleration/Deceleration, Performance, Altitude, and Ingestion test conditions.

Starting/Shutdown tests are done to test the engine during throttle transients that increase engine speed (called "spool up/spool down") to verify both internal dynamic component and starter performance.

They are usually performed at standard, cold and hot conditions for a specified number of cycles (MIL-E-8593A, p 111). Transient tests are power spike or droop tests done to verify engine performance during brief low--high--low or high--low--high load conditions. Acceleration/Deceleration tests are done to verify engine behavior during a low to high or high to low load condition. Performance testing is done to verify power output at constant power settings, such as 75%, 90%, 100%, 110% of full power etc. Steady State/Fuel Consumption tests are a subset of performance testing and are performed at a power or RPM setting for long periods of time to verify endurance, fuel consumption, and engine component wear characteristics. Altitude Tests are performed in an altitude simulating test cell to verify engine performance at the required pressure altitude.

Ingestion Tests are those conducted wherein the engine has some foreign matter introduced into its air intake. Typical ingestion tests are icing (both sheet and pellet), sand, atmospheric water, corrosive material, armament gas, or bird carcass ingestion. Figure 20 shows water atomizers for use in icing tests.

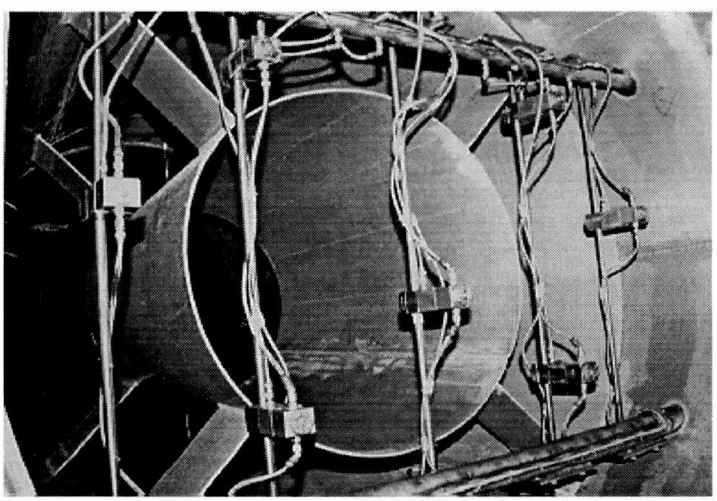


Figure 20 Water atomizers for icing tests

Test Standardization

In the course of this study, test conditions have been the hardest item to standardize. Each facility/engineer/test cell has its unique testing procedure peculiarities that must be taken into account

before a "generic" test is defined. For standardization purposes, the testing criteria for turboshaft engines found in MIL-E-8593A chapter 4 was used as a baseline. For a Qualification Test, MIL-E-8593A, chapter 4 outlines these procedures:

Endurance Test: Prior to testing, calibration of all control and sensor systems is required. The engine should be weighed and a bleed air sample collected from all ports at maximum continuous power. The engine shall then be subjected to an endurance test consisting of 25 cycles of 6 hours each, consisting of:

Maximum - Idle Run: 6 periods of 10 minutes each; 5 minutes of each period at maximum test power followed by 5 minutes at idle. On 5th, 10th, 15th, 20th and 25th cycles the power control will be in the manual mode for the first 3 minutes of the 5 minute maximum power condition.

Incremental Torque Run: 96 minutes total, in 8 and 4 minute increments, alternating between maximum and minimum output shaft speed and varying torque levels for each test condition. At the option of the user, 48 minutes may be replaced by alternating between the engines critical vibration conditions (these are vibration conditions that have been pre-determined to cause some sympathetic resonance in the engine components).

Power Transient Run: 39 minutes total. 20 minutes of varying between 4 minutes at no load, followed by one minute at maximum test power. 19 minutes of varying between one minute at maximum test power, decreasing immediately to a no load condition. As soon as the no load condition is reached, the engine will be immediately accelerated to the previous maximum test power condition.

Reverse Pitch Operation: 9 minutes total; one minute idle, 3 minutes at maximum reverse. If the engine does not provide reverse, substitute intermediate test power.

Intermediate Run: 30 minutes total; 15 minutes at intermediate power at maximum allowable shaft speed, and 15 minutes at intermediate power and maximum allowable torque. During the 5th, 10th, 15th, 20 and 25th cycles, the first 26 minutes of this test shall be run with the power control in the manual mode.

Maximum Continuous Run: 20 minutes total; 10 minutes at maximum continuous test power at maximum allowable output shaft speed, and 10 minutes maximum continuous test power at maximum allowable torque.

Intermediate Run: 15 minutes at intermediate test power, intermediate rate output shaft speed.

Maximum Continuous Run: 15 minutes maximum continuous test power at maximum continuous rated output shaft speed.

Intermediate-Maximum Run: 30 Minutes total; alternating three times between 5 minutes at intermediate test power at intermediate rated output shaft speed and 5 minutes at maximum test power at maximum rated output shaft speed.

Maximum Continuous Run: 16 minutes at maximum continuous test power at maximum continuous rated output shaft speed.

Intermediate Maximum Run: 30 minutes total; 15 minutes at intermediate test power and intermediate rated output shaft speed, and 15 minutes at maximum test power at maximum rated output shaft speed. When maximum power is limited to less than 15 minutes duration, maximum test power shall be run for that time duration and the remaining time shall be at intermediate test power. At 5 minute intervals the anti-icing controls will be operated at maximum for one minute.

Starting: A minimum of 300 starts shall be made during each 150 hour segment, on the endurance test engine. In addition, 10 false starts (spool and shut down) and 10 restarts (start immediately after shutdown) shall be performed. The engine shall be shut down and restarted not less than 6 times during each test cycle. Additional starts bringing the total to 300 may be made at the end of the endurance run. During these additional starts, immediately after the engine has reached idle speed, an acceleration to maximum shall be accomplished by moving the engine power lever within 0.5 second from idle to maximum power lever position with the power absorber loaded. The engine shall be held at maximum power for 30 seconds, then immediately shut down. The time between shutdown and starting for the first 18 of additional starts shall increase by 5 minutes from 5 to 90 minutes, then be fixed at 45 minutes for the remaining starts. Figure 21 shows an engine undergoing starting testing.

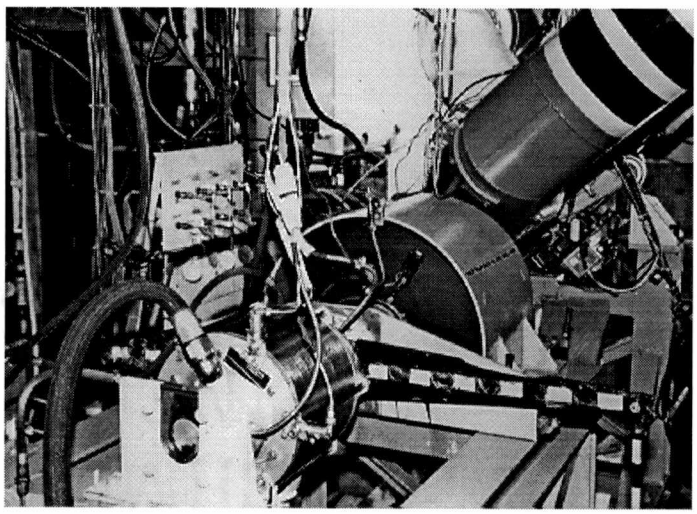


Figure 21 Allison T53-A-720 mated to a AVCO/Lycoming water brake during starting testing

Recalibration and Retest: The engine shall then undergo a recalibration to quantify any system degradation, then a total tear down analysis, reassembly and retest (MIL-E-8593A, p 112).

Altitude and ingestion tests are performed in their respective environments, but still utilize the basic procedures of a qualification test -- albeit in a shortened form. Performance testing is a series of long runs at constant rpm and power setting, frequently lasting 10 hours or more. Obviously, a single type of power absorbing device is not applicable to all the different kind of tests included in the broader category of qualification test. While it may not be possible to generalize such a device, it is certainly possible to restrict to a certain type of device for a particular engine test. By their very nature, certain test runs do not lend themselves to a particular power absorbing device. For example, due to reduced fluid density, an air dynamometer should not be used for an altitude test inside the altitude cell; the device could become unstable and would not provide the necessary inertia characteristic. This will be addressed further in the analysis.

CHAPTER V

TURBOSHAFT ENGINE APPLICATIONS

The last variable that must be taken into account when selecting power absorbers is the engine application. Rotorcraft demand a different type of schedule performance from an engine, as opposed to fixed wing aircraft, as opposed to armored vehicles or even watercraft. Each application carries with it a particular "critical modeling condition" that is the single most important portion of the operating envelope that must be accurately modeled in order to realistically test the engine. The inertia characteristics of the horsepower absorber used, coupled with its operating envelope and control system, dictate its applicability to a given test.

Rotorcraft

For rotorcraft the critical modeling condition is the low--high--low power transient that models a power-off maneuver simulating a forced landing, called an autorotation. During this maneuver, the pilot simulates an engine failure in flight by retarding the throttles to idle, and descending rapidly. If the maneuver must be terminated prior to touchdown because of an unacceptable flight condition (too slow, low rotor rpm etc) the pilot must accelerate the engines from idle to 100% operating rpm, then demand an almost immediate maximum power condition in order to arrest his rate of decent. The pilot then either lands the aircraft, or continues at a power condition other than maximum. This entire maneuver takes approximately 3 to 5 seconds. Figure 22 shows such a demanded power curve for this maneuver.

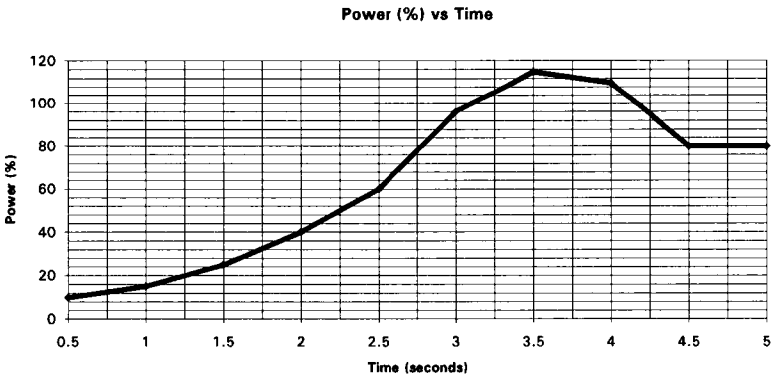


Figure 22 Power vs time for refused simulated forced landing
From: George, Marie, USAATCOM. Telephone Interview. 12 February 1996

Fixed Wing Aircraft

For fixed wing aircraft, the critical modeling condition is the single side acceleration/deceleration condition. This power vs time condition occurs when the pilot accelerates from idle to a maximum power condition for take off, or decelerates from maximum power to idle for a refused take off. Each maneuver occurs within 7 to 10 seconds, and consists of a rapid loading, followed by a steady state load. If taken sequentially, both must occur within 10 to 15 seconds, thereby simulating the power up, roll, refuse, and decelerate phases of a refused take off. Figures 23 and 24 illustrate these demand curves respectively.

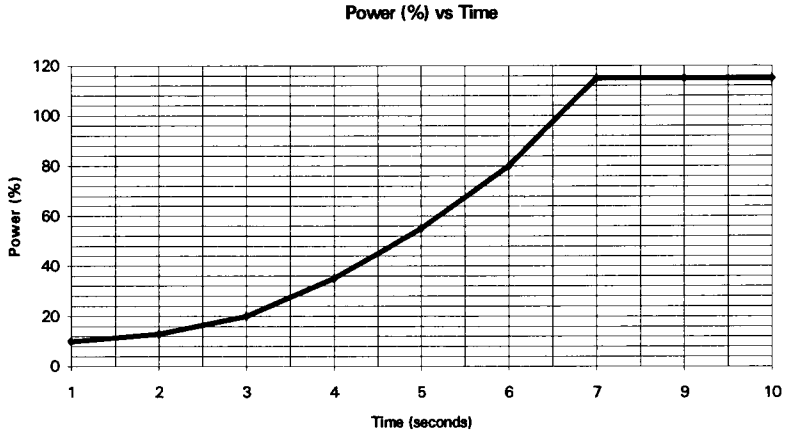


Figure 23 Power vs time curve for acceleration for fixed wing take off
From: George, Marie, USAATCOM. Telephone Interview. 12 February 1996

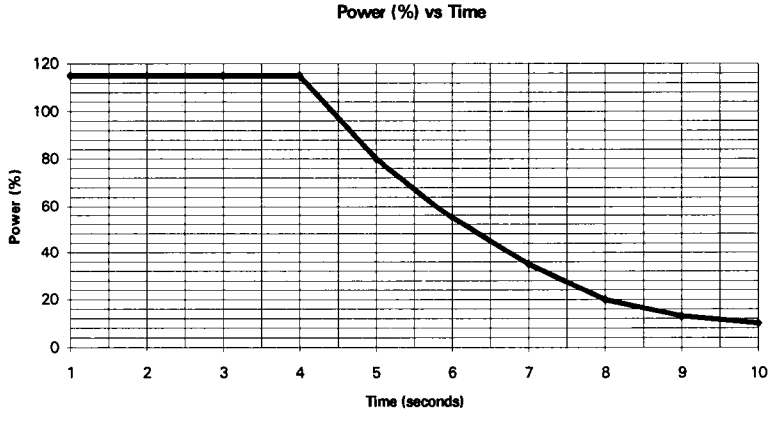


Figure 24 Power vs time curve for refused fixed wing take-off
From: George, Marie, USAATCOM. Telephone Interview. 12 February 1996

Armored Combat Vehicles

For armored combat vehicles the critical modeling condition is steady state torque and fuel consumption. Because the US Army's M1A1 main battle tank weighs 70 tons, and has very poor fuel economy, they are interested in how to squeeze more miles per gallon, at 45 mph, out of it's Lycoming engine. As such, testing done on these engines is development, not qualification. Any future testing would focus on specific fuel consumption or steady state performance, with transient performance a secondary consideration. Figure 25 shows a power demand curve for an armored combat vehicle test.

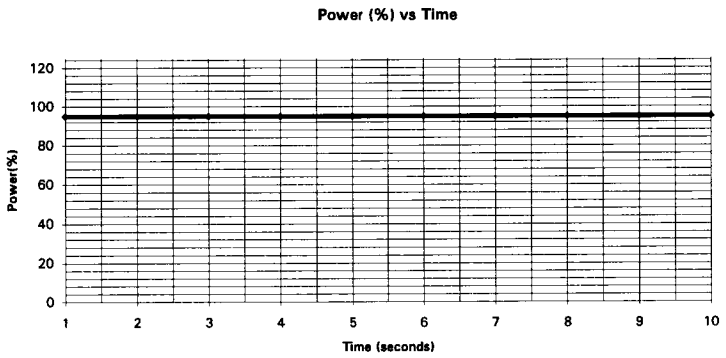


Figure 25 Power vs time curve for armored vehicles
From: Salter, William, USATACOM. Telephone Interview. 25 December 1995.

Watercraft

Finally, for watercraft, the critical modeling condition is the opposite of rotorcraft. Figure 26 illustrates the typical power demand curve for watercraft. Large racing boats use many different types of

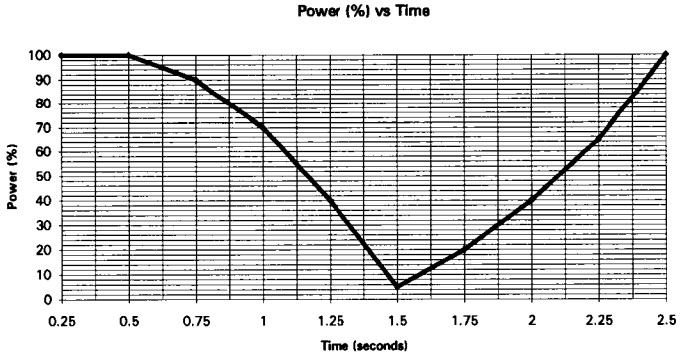


Figure 26 Watercraft power vs time
From: Voigt, Richard L. et al. Independent Turbine Testing and Research. New York: International Conference on Hydropower, 1987.

turboshaft engines, most in the 5000 shp level. The high--low--high power transient condition is important here, because the craft's propeller travels through areas of high to low to high fluid density as it moves into and out of water, and cavitates. The engine will move from a high load condition to a low load condition, then back to a high load condition in 1 to 2 seconds -- a very low time index. Most applications of this type have modified governor systems designed to throttle the engine down when it detects a low load, in order to prevent an engine overspeed; however, the system, by its very nature, has a built in acceleration lag. The engine response back to high power will not be immediate upon loading, but will lag behind 3 to 4 seconds to prevent surging (Baines, p 12)

Other turboshaft applications that were found, but are beyond this investigation, are: electrical power generation--both commercial and emergency, hovercraft, irrigation pumping, unmanned aerial vehicle propulsion, and aircraft auxiliary power units.

The reason it is most important to consider the critical modeling condition is that some power absorbing devices do not have the inertia characteristics and time responses necessary to accurately simulate the actual conditions under which the engine will operate. For example, due to its inherent instability during rapid deceleration, a water brake is probably not the best device to model a watercraft engine's high - low - high critical modeling condition. The device could become unstable once unloaded and either damage the engine, or produce useless data. A sophisticated control system could be introduced to steady the water brake, however, either an air or eddy current dynamometer would provide the necessary inertia and time index characteristics without major control system modifications.

The demanded power curves shown for the principle engine test described therefore play a role in the description of a matrix of recommended power absorbers versus engine test requirements that is presented in the following section.

CHAPTER VI

ANALYSIS AND RECOMMENDATIONS

The question we want to answer is: What is the best horsepower absorber for a given type test and test condition? The most important piece of information needed is the engine application. The engine application drives the "Critical Modeling Condition" with which we must concern ourselves. The critical modeling condition may eliminate one or possibly two devices. For example, when testing an engine used for rotorcraft, water brakes exhibit instability problems during rapid deceleration that frequently require elaborate control mechanisms to avoid failure. Because the critical modeling condition for rotorcraft requires both rapid loading and unloading, water brakes are probably not the best device to use. An air dynamometer is a possible candidate, however it has a very low inertial characteristic, causing the system to load or unload too quickly, not accurately modeling the load. The eddy current dynamometer is probably the most desirable device to use when modeling rotorcraft because it has a very well defined inertia curve, can accurately model the low--high--low power transient, and is stable at no load/low load high rpm conditions. It is also very controllable, reliable and has a predictable response to rapid loading. Air dynamometers are the second choice because of their low operating cost, and controllability as compared to water brakes.

For engines designed for fixed wing applications, eddy current dynamometers would also work well as power absorbers; however, similar results could also be obtained for sea level testing using air dynamometers. As previously discussed, acceleration/deceleration testing on fixed wing aircraft propulsion systems at sea level occur over short duration, of the order of a few seconds, followed by a steady state power condition. Air dynamometers can accurately model the inertia properties of propellers, are relatively inexpensive, are easy to control and provide repeatable results. Air dynamometers then, are the best choice.

From a modeling aspect, any device will serve for a power absorber to apply to tests of engines for armored vehicles. Therefore, for a steady state/fuel consumption test, either economy or test facility

configuration should be used as criteria to decide a best device. Water brakes may be best here because of their commonality, and good performance at constant loading.

Watercraft high--low--high transients can be best modeled and tested, as with the rotorcraft, with an eddy current dynamometer, for the same reasons. Water brakes and air dynamometers are not suited to high rpm and no load/low load conditions due to instability and tare loading. Idle tests (such a fixed wing aircraft operating at 100% operating rpm, but with no pitch in the propellers) are difficult for water brakes and air dynamometers to model without extra control systems. An eddy current dynamometer then, is the only device to accurately model the no load/low load conditions required for engines with these type of transient characteristics.

Once we decide on what critical modeling condition is important, and eliminated any devices, we must decide on the test condition and procedures, and thereby select the best power absorber. The test methodology and configuration must support the engines application and the test facilities capabilities. As an example, at NAWCAD Trenton, the 8W test cell is configured to simulate the entire dynamic componentry of the Sikorsky H-60 helicopter. All the dynamic components are mounted in scaffolding and operated at conditions similar to those encountered while in the airframe. The main rotor shaft is connected to a large gearbox to simulate the aircraft rotor system. Figure 27 illustrates the 8W gearbox. In this test condition, the gearbox height above the floor (over two stories), the small space requirements, and

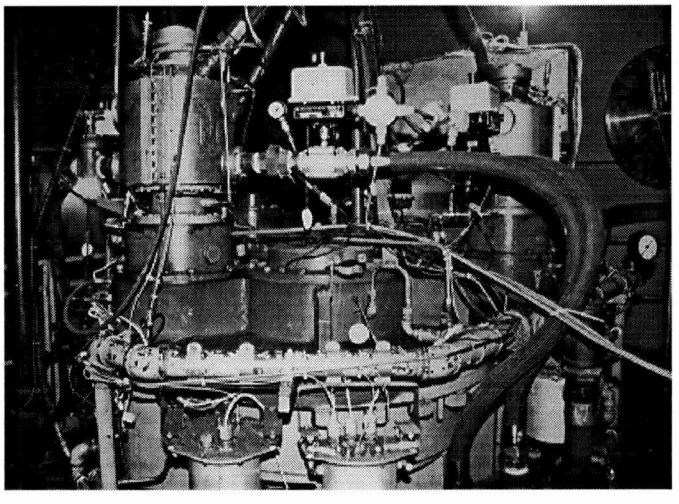


Figure 27 8W NAWCAD test cell gearbox -- showing multiple water brakes

a readily available treated water supply lead us to the conclusion that water brakes are ideally suited for horsepower absorption. Both eddy current and air dynamometers would be too large for this application, and further gearboxing would be impractical.

As another example, consider that an air dynamometer is not suited for an altitude test if installed inside an altitude chamber because the reduced air density would reduce the efficiency of the radial outflow compressor, thereby risking overspeed. Also, any device inside an ingestion test chamber obviously interferes with the foreign material trajectory into the engine inlet, and risks skewing test data because of a non-uniform environment. Any ingestion test, therefore, should be conducted with an unobstructed inlet. As figure 28 shows, the air dynamometer has a large cross-sectional area with relation to the engine intake. In front shaft, front intake engines, this means moving the horsepower absorber outside of the test cell, and connecting it to the engine via a gearbox and shaft arrangement. Another solution, albeit providing poor flow quality, would be to introduce a shrouded, low diameter water brake into the airstream, upstream of the atomizers, if shafting is impractical.

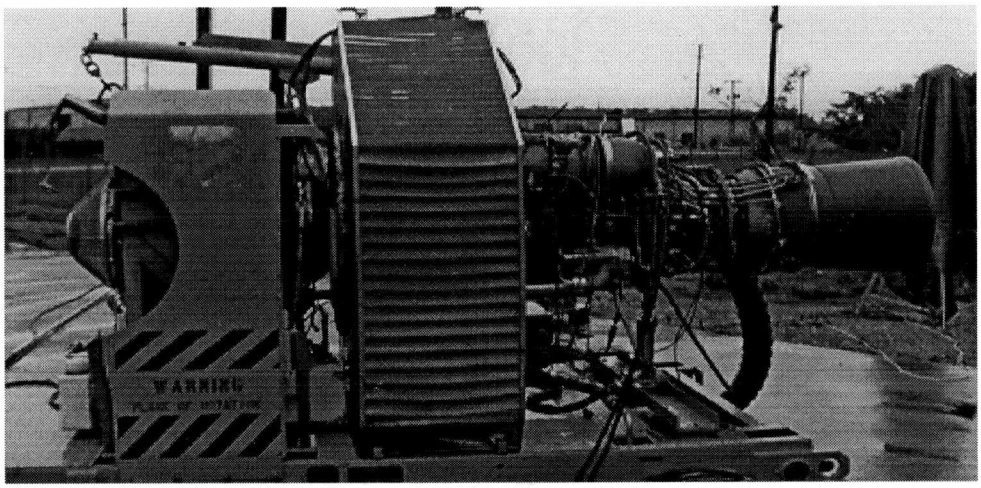


Figure 28 GE T700 Engine mated to a NREC 2500 BHP Air Dynamometer

For some types of tests, all of the power absorbing devices are equal , so we must concentrate on other discriminators to select the power absorber. Discriminators are: cost of operation, reliability, usability (ease of use), comfort level, and predictability. Comfort level, that level of familiarity with a particular device, should be considered in each decision. However, comfort level should not be allowed to

become the overriding factor if there are sound engineering and economic factors influencing another path. Both the US Army and US Navy have been using water brakes almost exclusively for the past 30 years in their turboshaft engine field facilities. While recently both services are using air dynamometers for limited testing, the majority of testing is still conducted with water brakes because of their long history. Figure 29 shows a 30 year old army test facility still in operation, using a water brake.

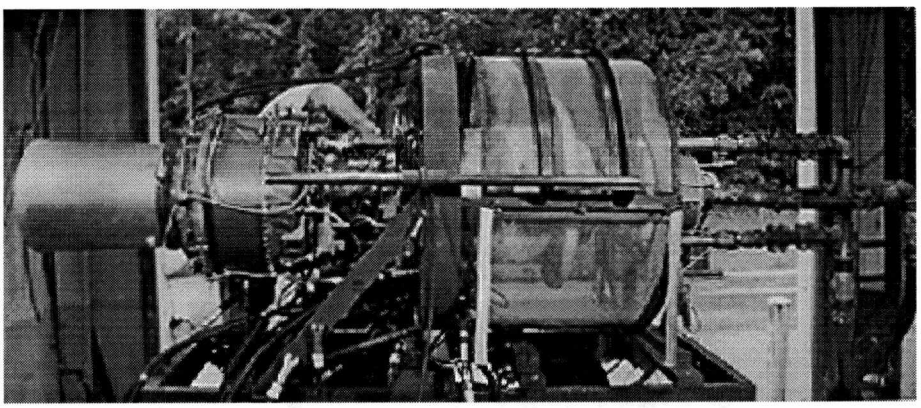


Figure 29 Lycoming T53-L-703 turboshaft engine mated to AVCO-Lycoming water brake in a Modular Engine Test Stand (METS) at Ft Campbell KY.

Clearly, each device has its operating trade-offs. Eddy current dynamometers are very versatile, but have high operating costs because of their water and electricity requirements, and have heat dissipation problems. Water brakes are less expensive, but have the same sort of support system requirements and can become unstable under certain loading conditions. Air dynamometers are simple, reliable, and do not require large support systems, but do not model some loading conditions, must be operated at or near sea level, and have a large cross-sectional area.

Considering the above trends, we can consider the best operating characteristics of each type of device, match those characteristics to the particular critical modeling condition, test condition and procedures for a particular test and choose a device to accurately model and dissipate the horsepower load.

A recommended "Test Matrix" for turboshaft testing, based on the qualification test procedures discussed in chapter 4 is given in Table 1.

	ROTORCRAFT	FIXED WING	ARMORED VEHICLES	WATERCRAFT
ENDURANCE TEST	AD is best for sea level testing because of low operating cost/high responsiveness. WB best for altitude testing because of low cross sectional area and cost advantage over EC . Shafting outside of test cell would allow any device.	AD is best for sea level testing because of low operating cost/high responsiveness. WB best for altitude testing because of low cross sectional area and cost advantage over EC . Shafting outside of test cell would allow any device.	Not concerned with much altitude testing; AD best for sea level because of low operating cost.	AD is best for sea level testing because of low operating cost/high responsiveness. WB best for altitude testing because of low cross sectional area and cost advantage over EC . Shafting outside of test cell would allow any device.
INGESTION TEST	When shafted outside of test cell, any device will serve; economics and comfort level dictate. When absorber must be inside test cell, WB because of low cross sectional area.	When shafted outside of test cell, any device will serve; economics and comfort level dictate. When absorber must be inside test cell, WB because of low cross sectional area.	When shafted outside of test cell, any device will serve; economics and comfort level dictate. When absorber must be inside test cell, WB because of low cross sectional area.	When shafted outside of test cell, any device will serve; economics and comfort level dictate. When absorber must be inside test cell, WB because of low cross sectional area.
STARTING/ SHUTDOWN	EC because of no load/low load modeling ability. WB and AD both unstable at no load/low load. Especially important if starting/shutdown is in conjunction with another test -- no change in device.	EC because of no load/low load modeling ability. WB and AD both unstable at no load/low load. Especially important if starting/shutdown is in conjunction with another test -- no change in device.	EC because of no load/low load modeling ability. WB and AD both unstable at no load/low load. Especially important if starting/shutdown is in conjunction with another test -- no change in device.	EC because of no load/low load modeling ability. WB and AD both unstable at no load/low load. Especially important if starting/shutdown is in conjunction with another test -- allows multiple tests.
TRANSIENT TESTS	EC because of ability to model entire operating range. Especially useful if performed in conjunction with another test -- allows multiple tests. WB if performed during ingestion with device inside test cell. WB will require digital control system	AD because of low inertia modeling. EC if performed at altitude and inside a cell. WB if performed during ingestion and inside the test cell.	EC because of ability to model entire operating range. Especially useful if performed in conjunction with another test -- allows multiple tests. WB if performed during ingestion with device inside test cell. WB will require digital control system	EC because of ability to model entire operating range. Especially useful if performed in conjunction with another test -- allows multiple tests. WB if performed during ingestion with device inside test cell. WB will require digital control system

EC = Eddy Current AD = Air Dynamometer WB = Water Brake

Table 1 Test matrix

If a single device had to be chosen: given enough resources, the eddy current dynamometer appears to be the most versatile horsepower absorber for general turboshaft engine testing. However, using a single device for an entire facility is not recommended.

Given the funding and facilities required, the ideal sea level test facility should have, as a general model, test facilities that are dedicated to each of the different types of horsepower absorbers so that multiple engines can be tested using multiple techniques simultaneously; a water brake facility, an eddy current facility, and an air dynamometer facility. Crossover capability could also be added to each, however, so that, if necessary, one could run an air dynamometer in the eddy current facility, for example. By operating different devices, each facility would use different resources, so trade-offs for electrical power or water filtration/cooling usage would be easier. This configuration provides inherent back up capability; some tests that should be conducted on a particular type of device, can often be conducted on another type of device with a small loss due to data incompatibility. This configuration would also provide the greatest flexibility for a facility because it allows continued, if restricted, operation of support systems, say cooling water, or large amounts of electrical power, become unavailable.

Altitude and ingestion facilities should be constructed to utilize any of the three types of devices for a given test. Electrical, air, and water support systems should be readily accessible at these facilities. Two test cells in close proximity to each other, sharing the same systems is probably the most economical configuration, but risks conflicts when both facilities need to use the same device. These facilities should be constructed with the horsepower absorber outside the test cell.

As a final note, data compatibility between devices should be addressed. The issue here is to determine if the data obtained from a test on a water brake is comparable to a test on an air dynamometer or comparable to a test on an eddy current dynamometer. Because the devices have different inertia characteristics and different control mechanisms they will usually produce different power vs. time curves. A rapid loading of an air dynamometer exhibits a steeper sloped power vs. time curve than the same loading of a water brake. Clearly the engine application should take priority, but when several types of devices are used to qualify an engine, can the data for one be used to predict the performance of another?

If the testing is primarily steady state, the answer is yes. If the testing includes transient tests, then, as of this writing, the applicable physics between the devices still requires analysis, and is beyond the investigation of this effort.

CHAPTER VII

CONCLUSION

After a historical perspective, each major type of horsepower absorbing device currently in general use was qualitatively analyzed for its power absorbing characteristics. Standards for turboshaft engine testing were discussed and a representative test selected and outlined. Turboshaft engine applications were discussed and critical modeling conditions analyzed. Based on the engine application, testing criteria, and device properties, a "Test Matrix" was developed recommending the "ideal" horsepower absorber to use for a given engine on a given test. Test facility configuration and data compatibility issues were briefly addressed.

As a final recommendation, the selection of a device as a power absorber can be made by considering the capabilities of the various devices, as provided in Table 1, and answering the following questions:

1) What is the engine application? (this drives "critical modeling condition")

2) What test are we doing and why?

3) What is the facility configuration?

4) Is the data compatible between test conditions and absorption devices?

   a. Does it need to be?

   b. What do we do if it is not?

5) What are the cost limitations imposed on the test program and will they restrict the acquisition and/or operation of the device?

APPENDIX

DIMENSIONAL ANALYSIS

Example Derivation of Power Absorber Performance Groups:

Power absorbers represent a unique class of machinery without well defined general theoretical performance models. This lack of general theoretical modeling makes it difficult to compare one type or size of power absorber to another type or size. It is not even clear which dimensionless groups control or correlate with fluid-based power absorber behavior. Because it is important that theoretical models for these devices be developed, a simple derivation is presented in this appendix as an example of an attempt to develop some of the characteristic groups of parameters that might correlate with device performance.

Power Dissipation Performance Group Derivation:

Because fluid-based power absorbers work by dissipating mechanical energy of the input power shaft by viscous fluid dissipation, the energy dissipation function for viscous fluids is the starting point. The function is restricted to two-dimensional flow, for simplicity.

$$\text{Dissipation Function} = \mu\Phi$$

Where:

$$\Phi = \left\{ \left(\frac{\delta u}{\delta y} + \frac{\delta v}{\delta x} \right)^2 + 2 \left[\left(\frac{\delta u}{\delta x} \right)^2 + \left(\frac{\delta v}{\delta y} \right)^2 \right] - \frac{2}{3} \left(\frac{\delta u}{\delta x} + \frac{\delta v}{\delta y} \right)^2 \right\} \qquad (1)$$

Assume that, in a strongly stirred flow:

$$\frac{\delta u}{\delta x} \approx \frac{\delta u}{\delta y} \approx \frac{\delta v}{\delta x} \approx \frac{\delta v}{\delta y} = \frac{V}{r_c} \qquad (2)$$

Where V is a representative stirring velocity and r_c is a representative length or scale of eddies in the stirred fluid.

Substituting (2) into (1) gives, after reduction:

$$\mu\Phi = \frac{10}{3}\,\mu\left(\frac{V}{r_c}\right)^2 \tag{3}$$

This function represents dissipated mechanical energy per unit volume of fluid, hence, for the entire power absorbing device:

$$P_{diss} = \frac{10}{3}\,\mu\left(\frac{V}{r_c}\right)^2 V \tag{4}$$

Where P_{diss} is the power dissipated by the device and V is the volume of the device.

Some simple assumptions for V and r_c are required. Assume that V represents a relative velocity, the difference between maximum rotor speed, and the mean fluid velocity of bulk flow through the device:

$$\text{define} \quad V = V_{rotor} - V_{fluid,bulk} \tag{5}$$

Hence,

$$V = \omega r - u_{bulk} \tag{6}$$

where:

$$u_{bulk} = \frac{\dot{m}}{\rho A_C} \tag{7}$$

A_c is a representative cross-sectional area for bulk fluid flow through the device.

By substitution; therefore,

$$P_{diss} = \frac{10}{3}\,\mu\left[\left(\omega r - \frac{\dot{m}}{\rho A_C}\right)^2\right]\frac{V}{r_c{}^2} \tag{8}$$

with

$$V = A_c L \tag{9}$$

and, assuming that r_c and L are functionally related, by, say a constant:

$$r_C = \lambda L \tag{10}$$

then

$$P_{diss} = \frac{10}{3}\,\mu\left[\left(\omega r - \frac{\dot{m}}{\rho A_c}\right)^2\right]\frac{A_c L}{(\lambda L)^2} \tag{11}$$

Finally, we can assume that the flow area and the rotor size are related, that, for simplicity

$$A_c = \pi r^2 \tag{12}$$

Then, by substitution

$$P_{diss} = \frac{10}{3}\,\mu\left[\left(\omega r - \frac{\dot{m}}{\rho A_c}\right)^2\right]\frac{\pi r^2 L}{(\lambda L)^2} \tag{13}$$

Hence,

$$P_{diss} = \frac{10}{3}\,\mu\left(\omega r - \frac{\dot{m}}{\rho A_c}\right)^2 \frac{\pi r^2}{\lambda^2 L} \tag{14}$$

Expand the squared term, and rearrange to give

$$P_{diss} = \frac{10}{3}\,\frac{\mu\pi}{\lambda^2}\left(\frac{\omega^2 r^4}{L} - \frac{2\omega r\,\dot{m}}{\rho\pi L} + \frac{\dot{m}^2}{\rho^2\pi^2 r^2 L}\right) \tag{15}$$

Because the constant λ is unknown, we rewrite equation (15) in terms of three constants, K_1, K_2, K'_3

$$P_{diss} = \mu\left(K_1\frac{\omega^2 r^4}{L} - K_2\frac{\omega r\,\dot{m}}{\rho\pi L} + K'_3\frac{\dot{m}^2}{\rho^2\pi^2 r^2 L}\right) \tag{16}$$

Noting that

$$\frac{\omega r\,\dot{m}}{\rho\pi L} = \frac{\omega r^3\,\dot{m}}{\rho\pi r^2 L} = \frac{\omega r^3\,\dot{m}}{\rho V} \tag{17}$$

With $m = \rho V$ being the mass of fluid in the power dissipater,

$$\frac{\omega r\,\dot{m}}{\rho\pi L} = \frac{\dot{m}}{m}\omega r^3 \tag{18}$$

The residence time for fluid to stay in the power dissipater is

$$\tau = \frac{m}{\dot{m}} \tag{19}$$

Hence,

$$\frac{\omega r \dot{m}}{\rho \pi L} = \frac{\omega r^3}{\tau} \tag{20}$$

Likewise, the term

$$\frac{m^2}{\rho^2 \pi^2 r^2 L} = \frac{m^2}{\rho \pi m} = \frac{\dot{m}}{\rho \pi \tau} \tag{21}$$

But,

$$\frac{\dot{m}}{\rho} = \dot{V} \tag{22}$$

Where $\dot{V}$ is the volumetric flow rate through the power dissipater.

Using equation (20) and (22) in equation (16) gives

$$P_{diss} = \mu \left(K_1 \frac{\omega^2 r^4}{L} - K_2 \frac{\omega r^3}{\tau} + K_3 \frac{\dot{V}}{\tau} \right) \tag{23}$$

where $K_3 = \frac{K'_3}{\pi}$.

Checking units, all of these terms represent Watts of dissipated power.

Equation (23) can be rearranged to the form

$$P_{diss} = \mu \frac{\omega^2 r^4}{L} \left(K_1 - K_2 \frac{L}{\omega r \tau} + K_3 \frac{\dot{V} L}{\tau \omega^2 r^4} \right) \tag{24}$$

From (24), we see that important characteristic groups for the viscous fluid-based power absorber:

$$\left\{\frac{\mu\omega^2 r^4}{L}, \frac{L}{\omega r\tau}, \frac{\dot{V}L}{\tau\omega^2 r^4}\right\} \tag{25}$$

We can examine these groups further.

The first group has units of power dissipation in watts, but, the other two groups are dimensionless. Clearly, from (25), the parameters of the problem or action of the power absorber are:

$$\mu,\ \omega,\ r,\ L,\ \frac{L}{r},\ \frac{\mu\omega^2 r^4}{L},\ \frac{L}{\omega r\tau},\ \text{and}\ \frac{\dot{V}L}{\tau\omega^2 r^4}$$

By conducting a test of a "standard" power absorber, it may be possible to correlate the performance or behavior of the device with these functional groups.

Reference 12 provides a standard dimensional analysis for viscous shear, axial feed water brakes. Reference 7 provides a cavitation driven dimensional analysis, and reference 16 provides a standard turbomachinery dimensional analysis. Some of these numbers are reproduced here:

Cavitation Number (Ca):	$\frac{P_s - P_v}{\frac{1}{2}\rho(\omega r^2)}$	Torque Coefficient (C_t):	$\frac{\tau}{\rho\omega^2 D^5}$
Reynolds Number (Re_r):	$\frac{\omega r^2}{\upsilon}$	Head Coefficient (C_H):	$\frac{H}{\omega^2 D^2}$
Flow (capacity) Coefficient (C_Q):	$\frac{Q}{\omega D^3}$	Power Coefficient (C_p):	$\frac{BHP}{\rho\omega^3 D^5}$
Viscosity Coefficient (C_m):	$\frac{\mu}{\rho\omega D^2}$	Thoma's Constant:	$\sigma = \frac{NPSH}{H}$
Roughness Ratio:	$\frac{\varepsilon}{D}$		

It may be that the approach illustrated here is not applicable to power absorbers. It was for this reason that it was termed an "example derivation." For this reason, additional theoretical modeling of power absorbers based on highly stirred fluids is recommended.

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VITA

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