DAMPER SEALS FOR GAS TURBINES AND COMPRESSORS

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Dr. Dara Childs and I have been testing seals for Texas A&M for about 15 years now. And our research has been aimed really at determining rotordynamic coefficients of seals with the idea of trying to minimize the vibrations of rotating machinery. And one of the things that we've learned is that seals have two major force coefficients that affect the vibration of the rotor. One of those coefficients is the direct damping, which simply opposes the whirl velocity. And the other coefficient is called the cross-coupled stiffness, which can create a force that drives the whirl and in some machines can cause a rotordynamic instability. And we've also learned that the damping coefficients are pretty much independent of the rotational speed. Whereas the destabilizing cross-coupled coefficients are very much dependent on the rotor speed because they're caused by the swirl of the gas.

We've tested conventional labyrinth seals in several different test rigs and found that they generally have no damping or sometimes negative damping. And they have high cross coupling, especially when the blades are on the rotor. Increasing the length of the laby seal in order to reduce the leakage increases the destabilizing coefficient. Dr. Childs extensively tested honeycomb seals with a smooth rotor and they have clearly a lower leakage than labyrinth seals. If you have a large length to diameter (L/D ratio) of honeycomb seals there's improved damping and small cross coupling.

I've invented a seal which we call the TAM seal. That's an acronym for Texas A&M seal. It has remarkably high damping and practically no cross coupled stiffness. It does leak more than a conventional laby seal but it works very well as a damper at all L/D ratios. I want to explain how the TAM seal works. Here what I'm showing with the dashed line is a shaft surrounded by a two bladed labyrinth that I have modified to make the TAM seal. I segment the circumferential cavity into four individual cavities with these walls. And I make the clearance of the inlet blade smaller than the clearance of the exit blade. What I mean by inlet and exit is upstream and downstream of the seal. In the drawing the flow is from left to right and so the vibration or the whirling of the rotor will modulate the flow into the cavities because the inlet clearance is changing with the vibration. But the flow out of the cavity will be modulated less because the exit clearance is larger. And it turns out now that the pressure in the cavities will modulate or oscillate at a frequency equal to the shaft vibration and the pressure in the cavities will be always opposing the vibrating velocity or the whirl velocity. The mathematical analysis shows that ideally you would like for the exit clearance not to be modulated by the shaft motion at all. And so I've been trying to think of different ways to do that. I want the flow into the cavity to be modulated by the shaft motion but I want the exit flow not to be modulated. One way to do that is with a brush at the exit, which I will talk about later.

I've tested these seals, my graduate students have tested these seals in both rotating rigs and non-rotating rigs. Here is a schematic of a rotating rig with the seal journal overhung. On each side of the pressure supply, there's actually two seals. The air comes in between two seals and blows out in opposite directions so here we can have two TAM seals. We've also tested conventional seals in the same test rig. Here's a photograph of that test rig. Here's the seal journal out here. And we have an air turbine augmenting the electric motor to get rapid accelerations when we need that. And then we also have a non-rotating rig that allows us to get some very rapid test results for just the damping when we're not interested in the swirl. Here we have the seal journal mounted at the top of cantilever beam and we just pop it with a hammer and let it vibrate freely and the air blows out the clearance around the seal. We get remarkable amounts of damping from the TAM seal, as you can see. At the top of the slide we see a non-rotating test. Here is the free vibration decay of a conventional laby seal and here is the TAM seal with the same dimensions on the same test rig.

On the rotating test rig here is the Bode plot. This is the synchronous response to imbalance of the rotor as it comes down from about 6,000 rpm to 0 and here is the base line with no air going through any seal. Here's the response with the conventional laby seal and here's the response with the TAM seal. And since this test was done we have doubled the damping of the TAM seal. Here are some summaries of tests. At the top of the slide we have the maximum amplitudes from the Bode plots of each test. Here they are for the conventional seal and here's the TAM seal. This one is for measurements in the horizontal direction and the other for the vertical direction.

Now I've said that we have doubled the damping of the TAM seal. The way we did that was to make the blade clearances the same, but put notches in the exit blade. We made it with the same clearance but we put little slots in it so that the flow out of the cavity at the exit blade won't be modulated by the shaft motion. Now it has occurred to me that a brush seal will do the same thing and allow perhaps a lower leakage. We've put this seal in a number of compressors, high pressure compressors. These are machines that make a lot of money in the oil industry and when they have a subsynchronous instability they can not be run and so they're very anxious to make it be able to run. And so they're not so interested in the leakage, they mainly just want it to be able to run. So here's a machine that wouldn't run because of the violent instability and we completely suppressed that with the TAM seal. Here's a waterfall plot showing the subsynchronous instability and we, I'm going to skip a lot of overheads here and show you the bottom line. Here is a spectrum after they put squeeze film dampers in the machine and the subsynchronous vibration is reduced but it is still there and with the TAM seal it was all but completely eliminated. It requires a magnified scale to even see the subsynchronous vibration with the TAM seal. This case illustrates the advantage that a seal can have as a damper because the seal is located in a better place to do damping on rotordynamic modes. Here's the mode shape of that compressor with the bearings out here at the nodes so that whatever damping you put into the bearings isn't going to do much good. But the center seal is located right here where the big amplitude is. And so it turns out that a damper seal will work X times better than a bearing damper, where X is the square of the ratio of amplitudes. So, in other words, if your seal has twice the amplitude of your bearing in the mode shape, it's going to work four times better than the bearing as a damper. I talked to a number of engineers and engine manufactures about using this in aircraft engines and I've run some cases that they gave me. I found that the TAM seal will generally have more damping than a hot squeeze damper in military engines. And it's located in a much better place, but they don't like the higher leakage. And so what we're hoping to do now is replace the exit blade with a brush seal which has very low leakage. But the TAM seal doesn't damp vibration unless it has some leakage. So what we want to do is active control. We want to have a deal where the normal operating condition is using the brush seal to produce a low leakage and then if we sense that we have a high vibration or if we sense that we have a blade rub, we'll open some ports and vent the flow out and produce the high damping just when we need it. And I think it will be really nice if we can combine that with a blade rub detector. So we're working on that also. We have this blade rub test rig in our laboratory. It is a 9,000 rpm axial flow blower with a housing that we can move back and forth to create blade rubs. We've been measuring the vibration signature that's produced by blade rubs and I'm working with another company in Austin called SPEC. We're trying to identify a special signature that will allow us to tell when a blade rub occurs. Then we can open the ports of the damper seal and get large damping right when and where we need it.

QUESTIONS

Q. What radial clearance typically do you use on your fitting blade on the current TAM seal?
A. We're doing about generally 1 mil per inch of diameter. Sometimes it varies. Right now we're doing testing on 4 inch diameter and we have 4 1/2 mils.

Q. Is there concern, John, with touchdown of those blades against the shaft or do you normally design large enough that you don't have touchdown?
A. Well, if you are thinking of wearing the seal away, that's going to deteriorate the performance of the damper because the damping goes up as the clearance goes down.

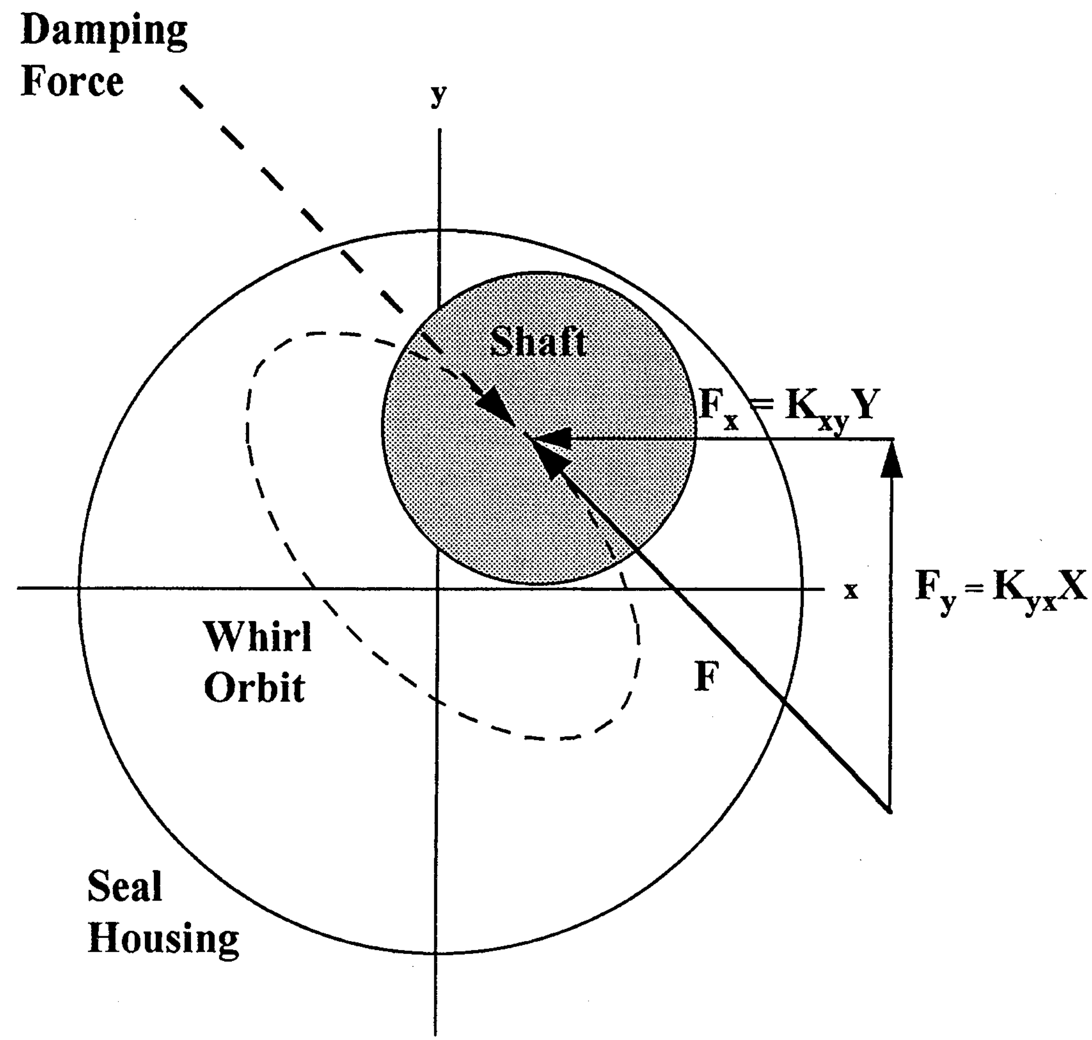
Q. And conversely, possibly wear or cutting of the shaft for the rotor.?
A. Well, of course at the inlet blade we just have a standard laby. So it's the same concern as you have with the laby seal. It's just a blade and you have the same concern that you have for the laby seal. On the brush, it's the same problem as with an ordinary brush seal.

Q. Well, normally configured for critical applications to put the blades on the shaft and let them wear into the stator, less stress that way.
A. That's right. This is, so far, these seals are strictly blade on stator and smooth rotor.

Q. John, since the seals depend upon leakage, will the clearance then also be dependent upon the pressure drop across the seal? Is there more pressure drop across the seal and more leakage as you adjust clearance?
A. Well, damping goes up with delta P and so far we've found a linear increase and we keep trying to get more pressure. We're setting up now to run 250 psi, but we find that damping goes up with delta P and damping goes up as clearance goes down until you get to a point and then there's an optimum where the damping starts to decrease, but that optimum is at such a small clearance that you couldn't run it.

ROTORDYNAMIC PROPERTIES OF ROTATING SEALS FOR GASES TESTED AT TEXAS A&M UNIVERSITY

- **Conventional labyrinth seal**
  Zero or negative damping, high K_{xy}, increasing length to reduce leakage increases K_{xy}
- **Honeycomb stator/smooth rotor**
  Low leakage and high damping at large L/D ratios
- **TAMSEAL**
  High damping, almost zero K_{xy}, higher leakage, works well at all L/D ratios
- **Brush seal**
  Lowest leakage, no damping, same K_{xy} as a labyrinth



The cross-coupling coefficients, K_{xy} and K_{yx}, produce the destabilizing force F. The TAMSEAL both minimizes F and increases the opposing damping force, thereby greatly reducing vibration.

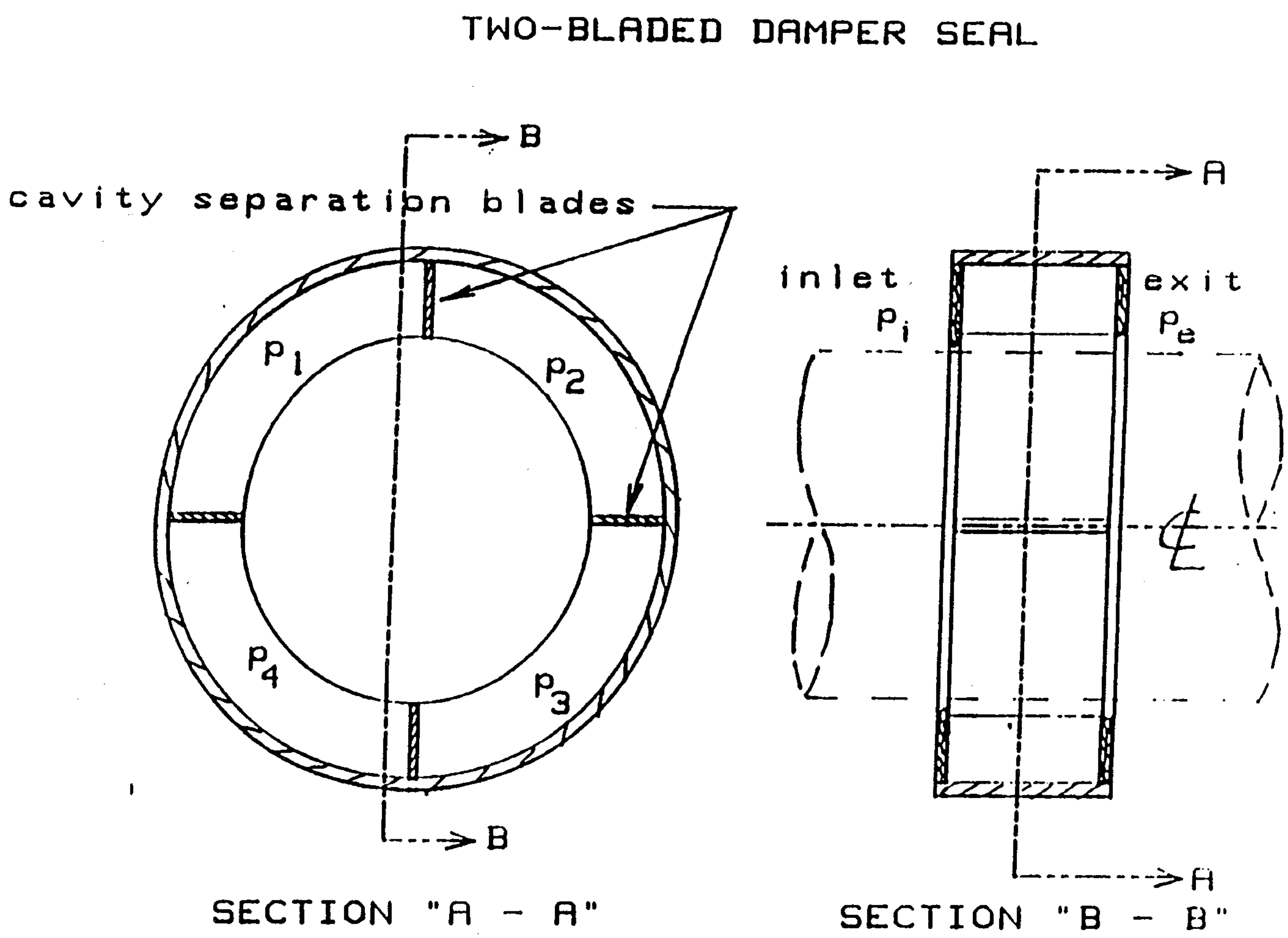


Figure 1. Two-Bladed Damper Seal

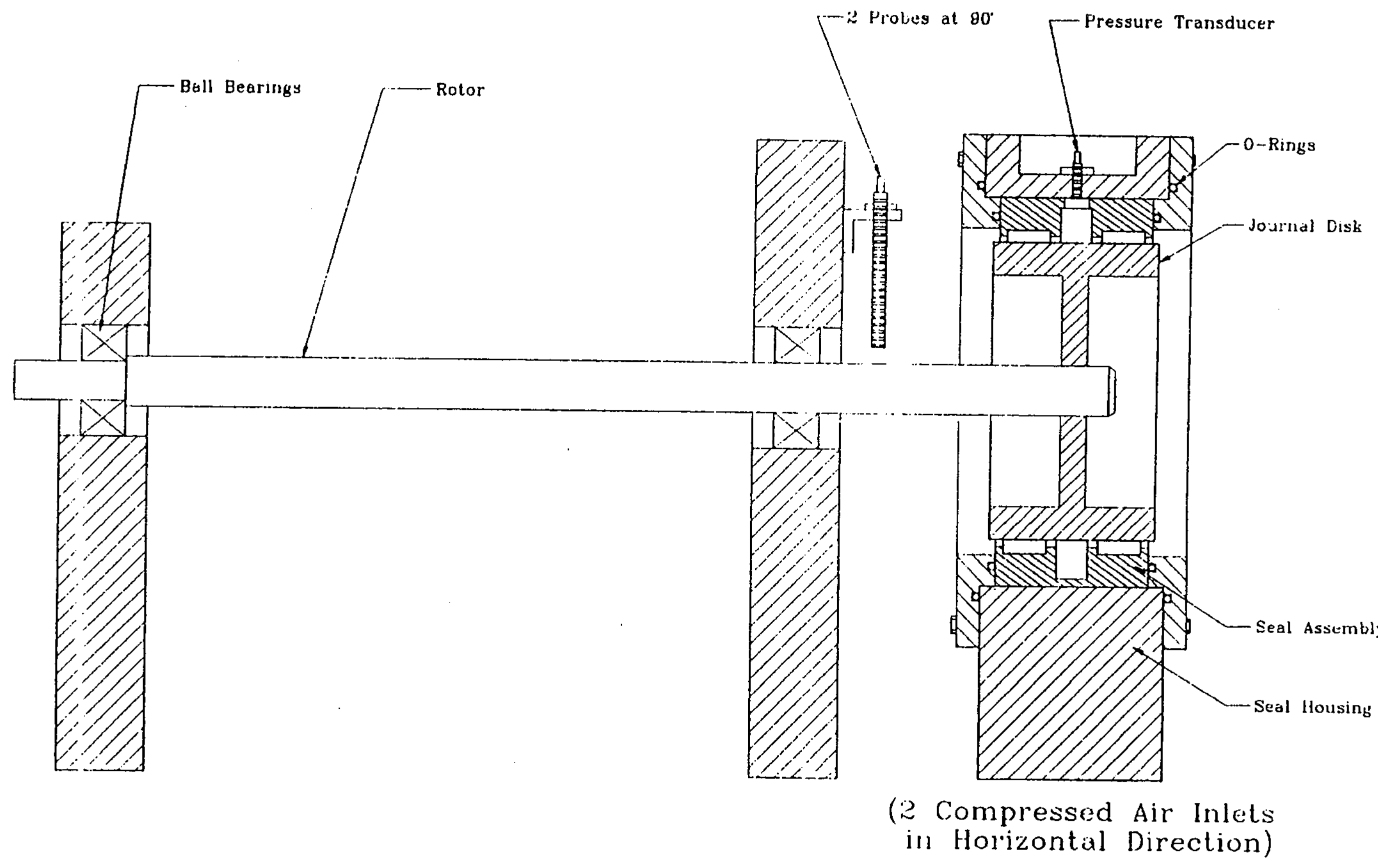
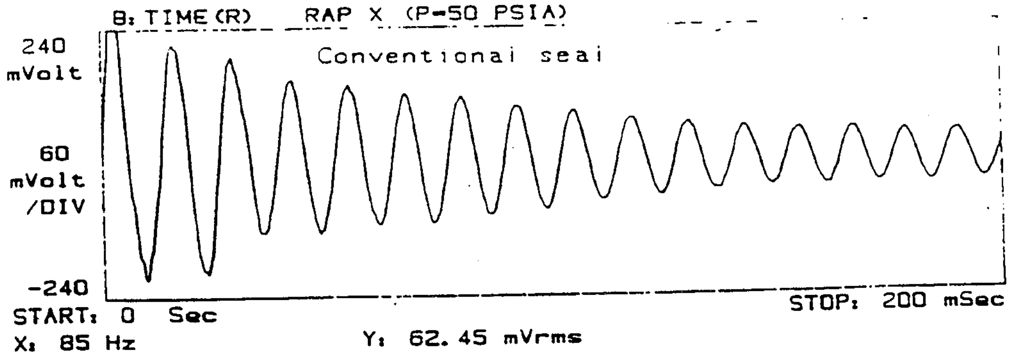
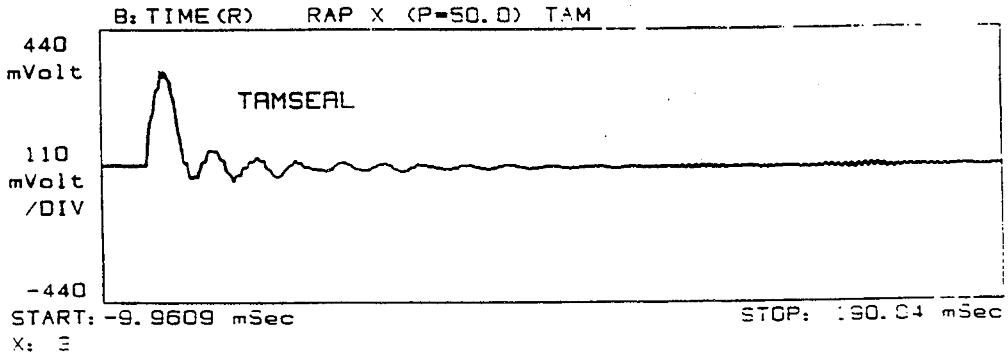
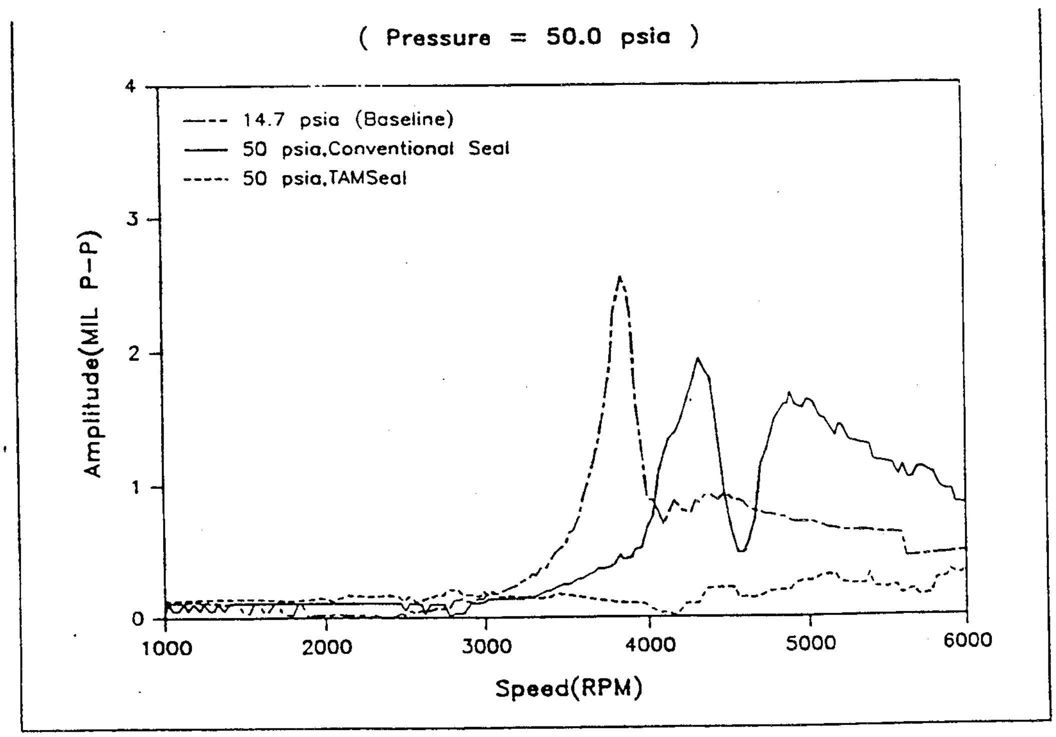


Figure 2. Cross-Section of Damper Seal Test Rig





Rap test results



Coastdown results

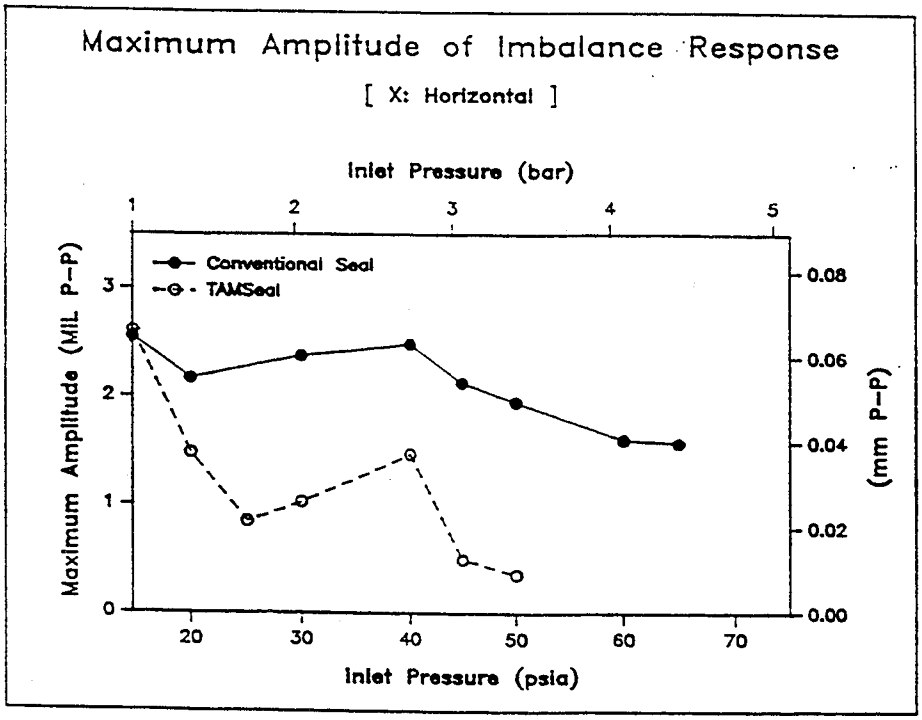


FIGURE 3. CRITICAL SPEED AMPLITUDE VS. INLET PRESSURE

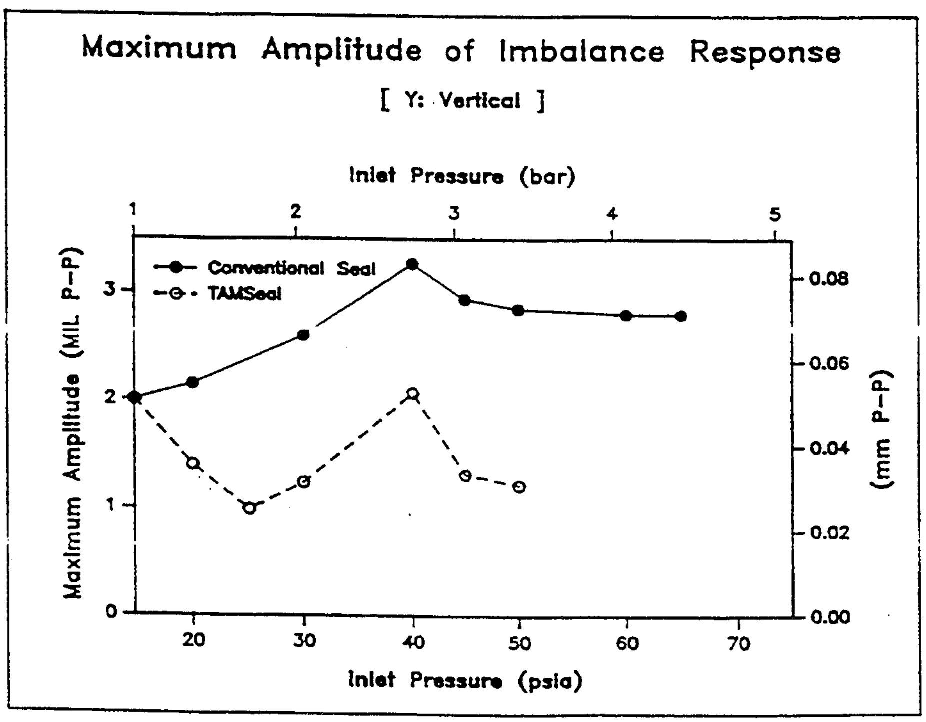
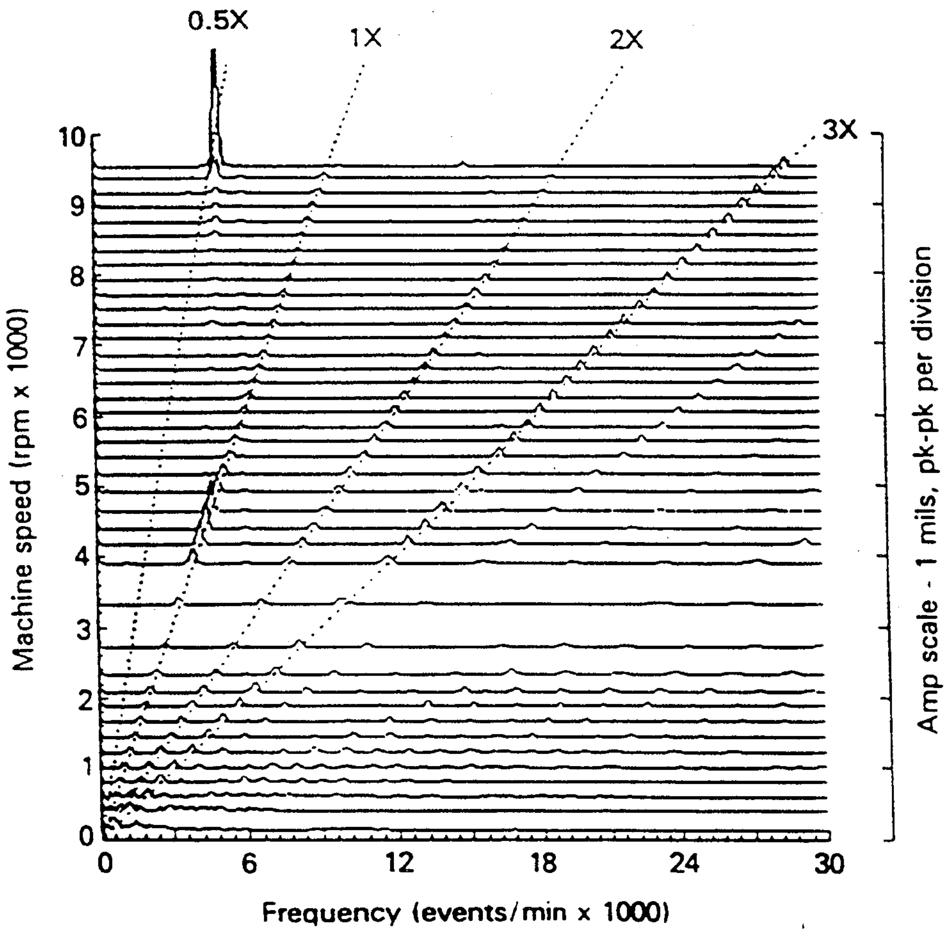


FIGURE 4. CRITICAL SPEED AMPLITUDE VS. INLET PRESSURE

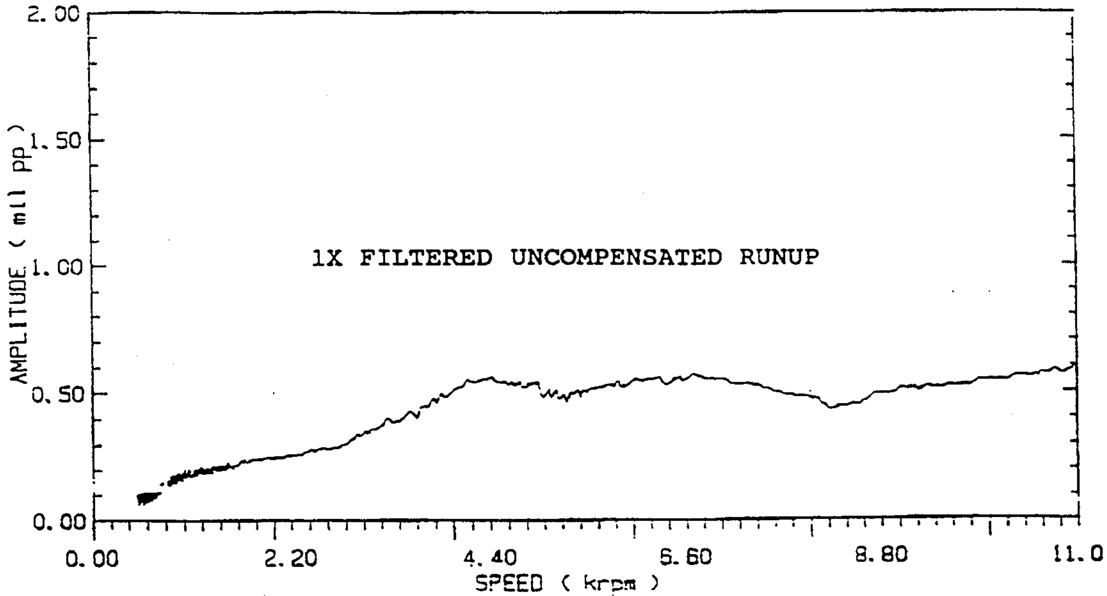


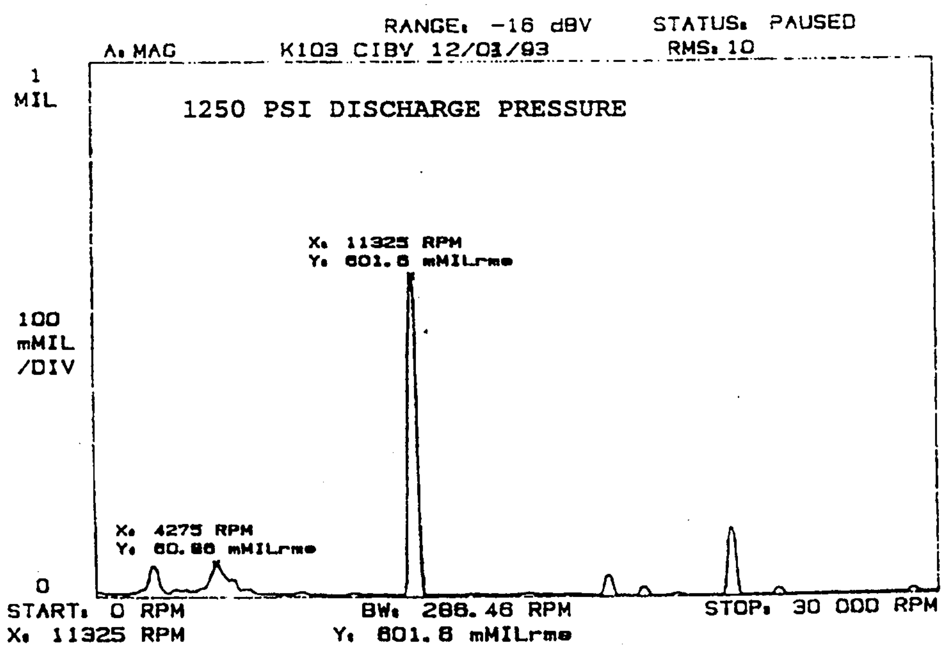
Waterfall Plot of B-1 Compressor Runup—Thrust End Vertical Probe.

"amplitude is 0.004 to 0.006 in, peak-to-peak resulting in an ESD. Subsequent examination of the labyrinths revealed a midspan excursion greater than 0.060 in. radial. At this time, a complete rotordynamic analysis was undertaken by in an attempt to identify the problem."

MEASURED VIBRATION OF THE SIX STAGE COMPRESSOR
WITH: ORIGINAL BEARINGS
DRY GAS SEALS
NO SQUEEZE FILM DAMPER

From Kocur et. al., Sixteenth Turbomachnery Symposium, 1987





MEASURED VIBRATION OF THE SIX STAGE COMPRESSOR
WITH: ORIGINAL BEARINGS
NO SQUEEZE FILM DAMPERS
TAMSEAL AT CENTER LABYRINTH

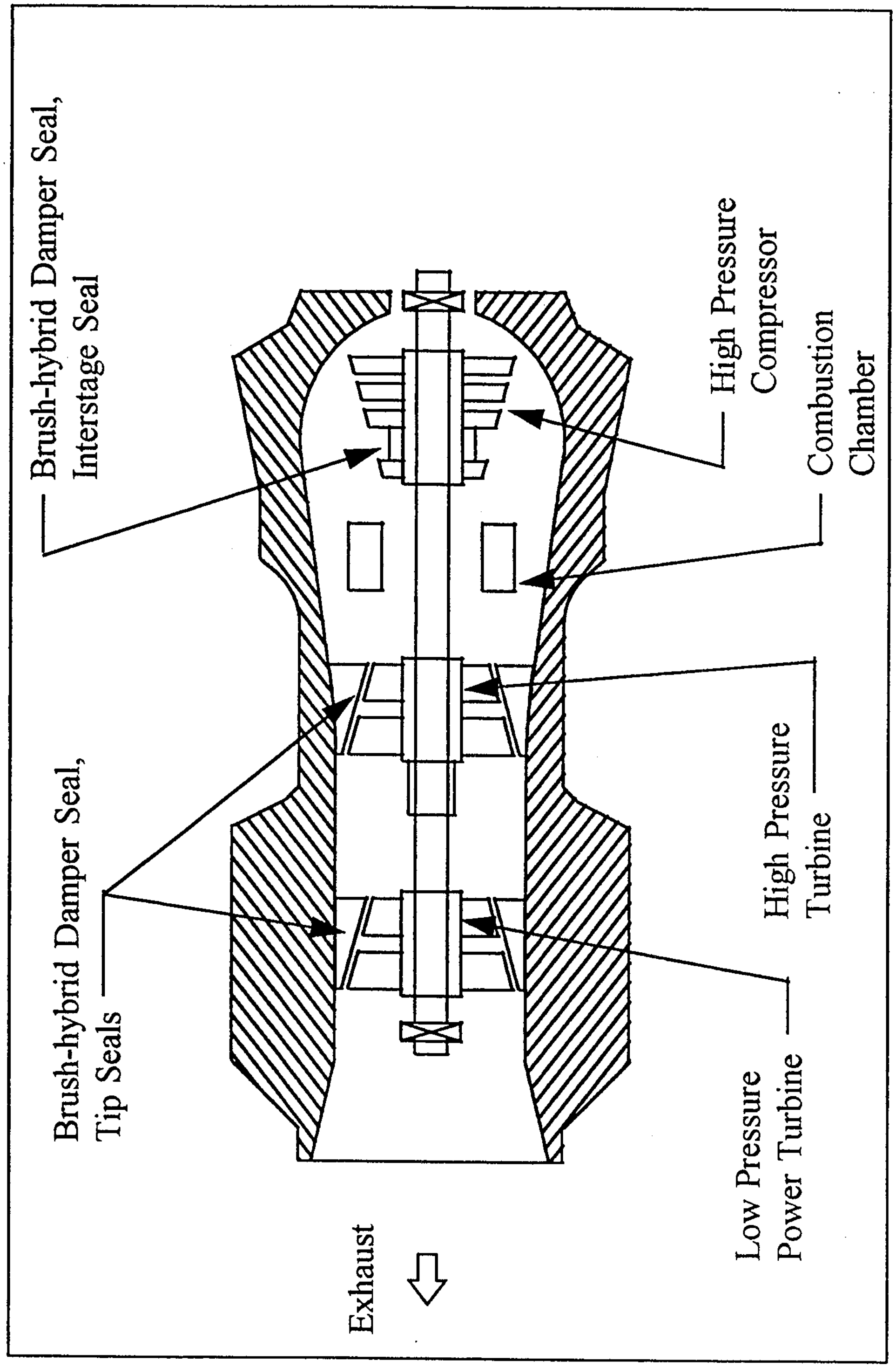
December 3, 1993 field test

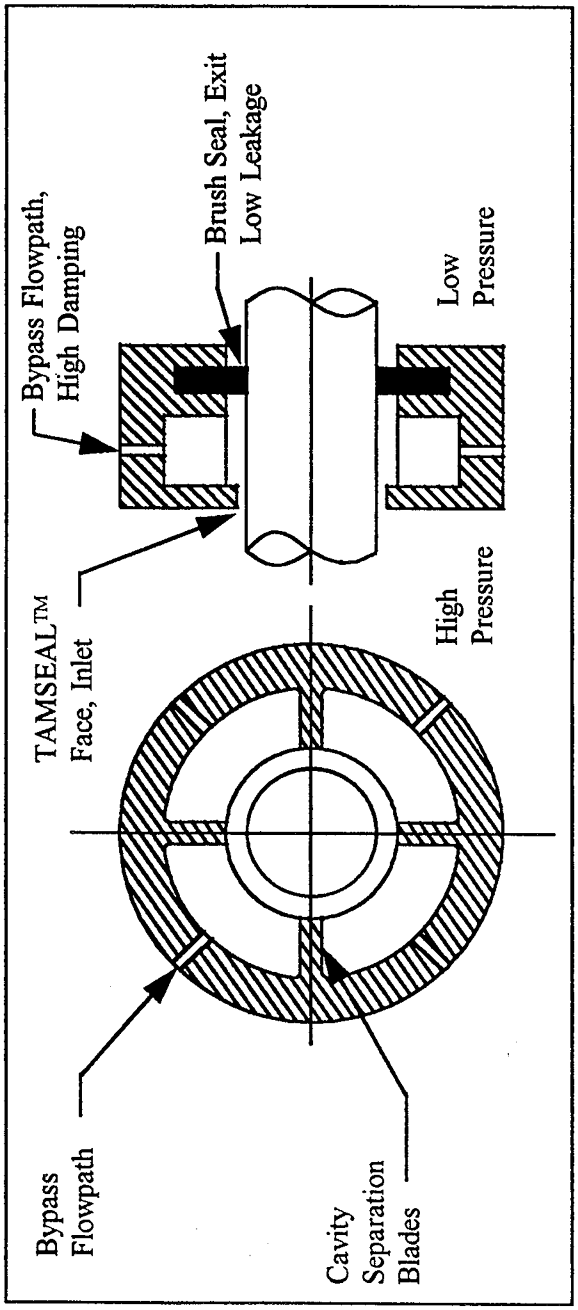
Seals are much better located for modal damping than bearings. Effective damper seals can minimize blade rubs, thus improving performance and efficiency over a longer engine life. The ideal seal for vibration damping in gas turbine engines is the TAMSEAL

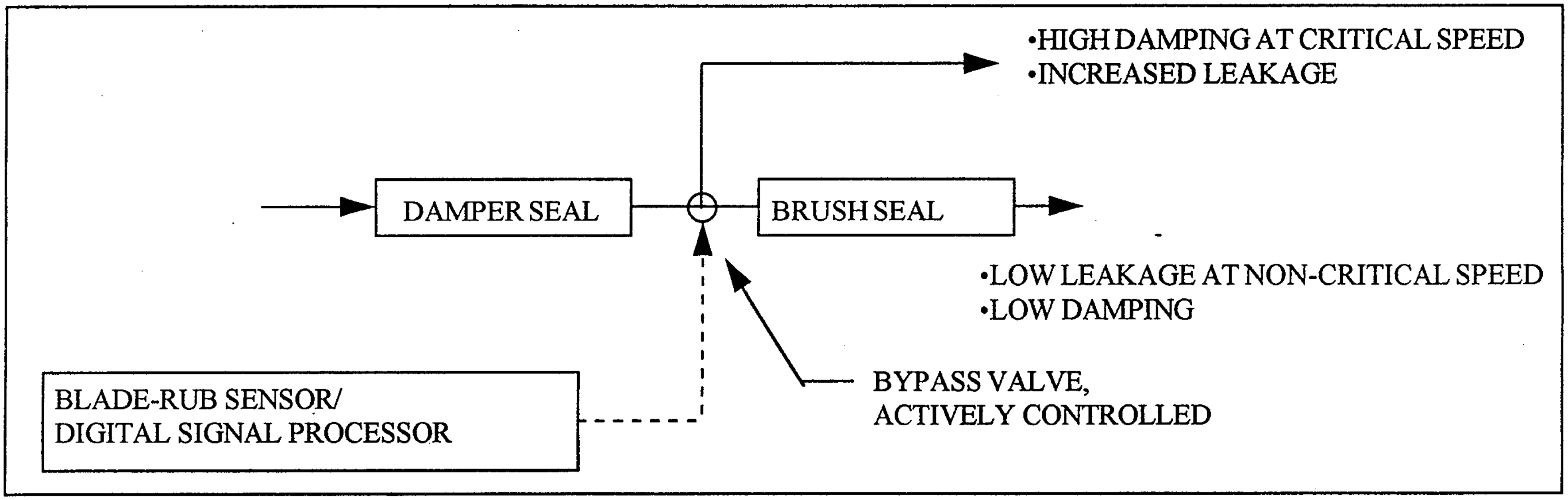
- Low axial space requirements
- Blocks swirl and minimizes K_{xy}

Problem: the TAMSEAL requires a certain amount of leakage to produce its damping forces. Low leakage brush seals have no damping capability.

Solution: Combine the TAMSEAL with a low leakage brush seal. Activate the TAMSEAL mode whenever a blade rub is sensed.


Brush-hybrid Damper Seal,
Interstage Seal
High Pressure
Compressor
Combustion
Chamber
Brush-hybrid Damper Seal,
Tip Seals
High Pressure
Turbine
Low Pressure
Power Turbine
Exhaust


Bypass Flowpath, High Damping
Brush Seal, Exit Low Leakage
TAMSEAL™ Face, Inlet
Low Pressure
High Pressure
Bypass Flowpath
Cavity Separation Blades



A TAMSEAL installed in series with a brush seal. A blade rub sensor controls the valve to bypass the brush seal which increases damping and reduces blade rub at critical speed.