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MAGNETIC SUSPENSION ACTUATOR CONCEPTS AND APPLICATIONS

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SUMMARY

The fundamental aspect which makes magnetic suspension systems possible is the magnetic phenomena by which significant forces can be generated. Each of these force-producing phenomena has unique characteristics and is implementable in a unique fashion, such that each performs the magnetic suspension task differently than the others.

This paper provides a practical overview of the force-producing concepts, their unique characteristics, and their typical methods of application.

INTRODUCTION

Magnetic suspension systems have become the focus of intense interest and development in recent years. Applications for these magnetic suspensions vary widely but may be grouped into two rather broad classifications.

The first classification would be a spinning shaft group of applications including momentum wheels, energy storage wheels and rims, and gimbals. In this application group, the magnetic suspension ideally duplicates the stiffness performance of the conventional ball or roller bearing and is substituted for the conventional bearing, either to achieve longer life due to the absence of contact wear or to eliminate the thermal conduction path inherent with direct contact. Although not spinning, reciprocating applications, such as cryogenic refrigerators, would also be included in this group, since performance equivalent to a linear roller bearing would be the ideal.

The second application group would be the suspended platforms, characterized by the desire to vibration isolate and/or point the payload or experiment on the platform with respect to a second (reference or stationary) body. In this group, the magnetic suspension is ideally very stiff or rigidly controlled in the low-frequency pointing region and soft in the frequency region where vibration isolation is desired.

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ACTUATOR CONCEPTS

This section will introduce and discuss the magnetic suspension actuator concepts which would be considered and could be utilized in normal ground and space environments (typically one atmosphere pressure to hard vacuum and a temperature range from -50 to +80°C). In a later section, alternate concepts and design modifications for cryogenic temperatures will be discussed.

The fundamental basis which makes magnetic suspension systems possible is the magnetic phenomenon where significant forces can be generated between mechanically isolated bodies. Each of these force-producing phenomena has unique characteristics and, therefore, has unique methods of implementation and areas of application.

The force concept, which is the basis for two distinctly different actuators, is the minimum reluctance principle. This principle states that, when a magnetic flux flows between two bodies separated by a gap, a force is generated between those bodies in a direction to reduce the gap reluctance.

One actuator, based on this concept, is the ferromagnetic attraction actuator. In its simplest form, it is a single horseshoe electromagnet operating against a magnetic steel armature plate (fig. 1(a)) and producing unidirectional force on the armature plate. The addition of a second electromagnet (stator), operating against the other side of the armature, creates an actuator capable of bidirectional force, with the stators controlled differentially (fig. 1(b)). This actuator is an active actuator since active control of the excitations to the two stators is required to obtain the desired force characteristics and to maintain stability.

The second type of actuator, based on the minimum reluctance principle, is the variable reluctance actuator in which a magnetic field passes between two bodies, with the gap surface of each body consisting of a matched series of teeth and grooves (fig. 2). The force generated is always in a direction to align the teeth on the movable element (armature) to those on the stator. The field may be generated by either permanent magnets or by a coil as in the electromagnet. The more common configuration uses a permanent magnet to create the field, resulting in a completely passive actuator; then, should a variable force actuator be desired, a coil is utilized to modulate the permanent magnet field.

A second force concept commonly employed in a vast array of magnetic devices is the Lorentz (or D'Arsonval) force principle. In this concept, a current-carrying conductor in and perpendicular to a magnetic field has a force exerted upon it which is perpendicular to both the field and the current (fig. 3(a)). Magnetic suspension actuators of this type are very similar in operation and construction to the common speaker actuator or voice coil (fig. 3(b)). This actuator obviously requires active control to obtain the force magnitude desired.

The third force concept that can be used is the repulsion force created between like poles of permanent magnets. When configured as a radial actuator (fig. 4), it produces a net radial force in a direction opposing any radial offset of the movable rotor element with respect to the stator. It is a passive actuator and performs in a manner similar to the variable reluctance actuator.

ACTUATOR ATTRIBUTES AND APPLICATIONS

The ideal magnetic suspension would provide all of its advantages: long life, no contact, no wear debris, and vibration isolation, with no penalty in weight, power or complexity compared to its mechanical counterpart. However, this is never achieved, and the magnetic suspension system and its actuators must be optimized to the application in order to be competitive with alternate devices, such as bearings, flexures, and dampers. Then, at the onset of magnetic system design, the requirements must be evaluated and a careful selection of the type of system, actuators, and control scheme must be performed.

Prior to discussing actuator attributes, several terms, which will be used in that discussion, require definition.

- Stiffness - will be used to denote the restoring force per unit deflection of the passive-type actuators. K_A will denote the axial stiffness of a passive axial actuator, and K_R will denote the radial stiffness of a passive radial actuator. The term "stiffness" will also be used for active actuators, based on measurable deflection levels, for comparison purposes.

- Unbalance Stiffness - will be used to denote the force per unit deflection, which acts in a direction to increase the deflection, such as in the cross-axis direction of the passive actuators. For example, K_{UA} will denote the axial unbalance stiffness of a passive radial actuator.

Table 1 provides a qualitative attribute summary for the four actuator types. These attributes are further explained and quantized in the following sections.

Active Actuators

The electromagnet actuator is applicable to both the spinning shaft and suspended platform application groups. In spinning shaft applications, where passive radial support is preferred for simplicity, the electromagnet actuator is used to provide control of the unstable axial axis. This can be accomplished by integrating it into the passive suspension, as shown in a simplified configuration in figure 5, or by incorporating it as a separate component. The electromagnet can be, and has been, used to provide the radial support in these applications. With high-resolution position sensors and an appropriate control system, it is capable of much higher radial stiffness than the passive actuators for a given weight and envelope. However, the added complexity of the control system for four degrees of freedom (DOF) (two radial and two angular) usually eliminates it from consideration.

The electromagnet actuator is the principal candidate for suspended platform applications. These payload pointing and vibration applications typically consist of five or six actuators, depending on the number of DOF to be controlled, located around or under the platform. The actuator is required to provide its full force capability (several newtons to several thousand newtons) over a significant motion range (a few millimeters to several centimeters). The force requirement usually decreases with increasing frequency but may be significant out to several thousand hertz.

The electromagnet actuator's competition for suspended platforms is the voice coil actuator. When traded off against the voice coil, one of the electromagnet's disadvantages is that its output force is nonlinear to both excitation current and gap flux density and varies as the inverse square of armature motion (gap), thus requiring complex control systems. Other weak points include its relatively high inductance (due to the iron core which can limit high frequency force generation or require very high voltage supplies), torques generated due to armature rotation (or tilt) about the two cross axes, and the fact that some power, although slight, is dissipated in the armature. On the other side of the trade-off, the electromagnet's passive armature, which is located on the suspended element, is light compared to the passive (magnet/return path) element of the voice coil; cross-axis translations are accommodated by the armature overlap where, in the voice coil, large magnetic gap clearances are required; and most important, for practically every set of force/motion requirements, the electromagnet is substantially lighter and lower in power than the voice coil.

The cross-axis torque produced by the electromagnet actuator becomes less of a problem as the actuator location radius increases, since the torques are generated as functions of the actuator pole spacing and pole face area and decrease in relative magnitude with increasing location radius. The force and torque errors of the electromagnet have been the subject of significant study and modelling at Honeywell in recent years, are reasonably well-understood, and are routinely incorporated into the system error budget from the onset of a suspension system design.

Configurations of the electromagnet that have been used in Honeywell suspension system designs include the horseshoe, as shown in figure 1(b), four horseshoes in a single component (similar to a motor stator), and the cup core consisting of a single coil with a pole in the coil center and a return pole around the coil outer diameter.

Sizing the electromagnet actuator is an iterative process based on the requirements flow-down from the system-level requirement set. Size, weight, and power increase with increasing force and gap motion requirements. They are also highly dependent on the type of feedback system employed (gap, force, gap flux and combinations) and the degree of required conformance to the force-proportional-to-current-squared and force-proportional-to-inverse-gap-squared relationships. Honeywell's more recent systems utilize gap flux density feedback, with the flux density squared closely approximating the actual force produced; thus, several error sources (hysteresis and current and gap square law errors), which previously required the use of low flux density core material and increased magnetic gap, are now eliminated. This allows the use of a high flux density core material, which reduces the core size and, since weight and power can be traded off against each other, results in a lower power and weight actuator.

The voice coil actuator has already been partially discussed in comparing it against the electromagnet actuator. It becomes a serious competitor only in applications where full force capability above approximately 20 Hz or significant forces above approximately 100 Hz are required or where extreme freedom from force and torque errors can offset the weight and power penalties incurred.

One application area to which the voice coil actuator is ideally suited is the control of fast steering mirrors where forces to 2000 Hz and higher may be required.

When selected for its low error capability, the configuration must be modified from the simple voice coil in figure 3(b), with magnets located only at the inner diameter of the gap. That configuration typically yields a gap flux density distribution, as shown in figure 6(a), and radial and angular coil motions in the gap result in force scale factor and cross-axis force/torque errors. Locating magnets at both the inner and outer diameters of the gap will result in the more balanced flux density distribution of figure 6(b) and thus minimize motion-induced errors.

Table 2 shows a portion of the trade-offs between active actuator types for a recent program. For this application, the electromagnet actuator, using the low-weight, high-flux density material, was the selected option primarily because of weight and the fact that the other options were 1.5 to 4 times more expensive to build.

Passive Actuators

As previously mentioned in the discussion of spinning shaft applications, passive radial suspension is preferred because of the simplicity it affords. With the two radial DOF and the two angular axes (orthogonal to the rotational axis) passive, the only penalties it incurs, relative to the ball-bearing system, are one DOF active control, added weight, and reduced stiffness. The advantages, of course, are the absence of contact and resultant wear particles, elimination of lubrication concerns, potential for longer life, constant drag level over life, and higher reliability.

The variable-reluctance passive actuator, in addition to a high weight-to-stiffness ratio (table 3), usually has a high unbalance stiffness (K_U)-to-radial stiffness (K_R) ratio. In most configurations, this K_U/K_R ratio is in the range of 5 to 10. This has disadvantages and advantages. Assuming a passive-radial, active-axial system, the high axial unbalance stiffness requires a very tight loop (high bandwidth) be closed on the axial axis, and stability is often difficult to achieve. The unbalance stiffness can be used to an advantage; in ground-based applications or in space applications where ground testing is required, it provides support for the weight of the suspended body in an axial 1g gravitational field. The technique uses the active axial actuator to offset the passive radial actuators axially to the point where the axial unbalance force equals the suspended weight; the active axial suspension then maintains this position, where its average power is almost zero. Because of problems caused by the high K_U, including larger active portions to provide initial liftoff from the stops, Honeywell has recently been evaluating alternate configurations whose K_U/K_R ratios are in the range of 0.5 to 2. Build and test of hardware based on this configuration will occur this year.

The other passive actuator, using permanent magnets operating in repulsion, is also being investigated. In the designs evaluated to date, this actuator suffers a weight penalty of almost 2 to 1, compared to the variable-reluctance type, and the K_U/K_R ratio is in the 1.0-to-1.5 range. A distinct disadvantage is that, with two magnetic fields in repulsion, the fields are very uncontained, which results in significant stray external fields. This is in comparison to all the previously discussed actuator types whose magnetic circuits are closed and whose fields are reasonably well-contained.

Table 3 provides a comparison of the four actuator types for an assumed rotating wheel application with a small motion range requirement. This example is chosen to demonstrate an application where the active and passive concepts are reasonably competitive and the choice is not obvious but is more related to desired performance.

ACTUATORS AND CRYOGENIC APPLICATIONS

Current investigations of cryogenic applications and the impact of using actuators at or near cryogenic temperatures lead to the following preliminary conclusions.

- Core materials, in general, do not suffer serious degradation at cold temperatures, although there are some notable exceptions, such as 49 percent nickel iron which reportedly loses about 45 percent of its permeability at -150°C and 75 percent of its permeability at 40°K.
- Impregnants usually used for the 25 ± 75°C range are not applicable to cold temperatures, but alternate materials exist.
- Power consumption in active actuators can be reduced to near zero, using superconducting cable, but the complexity of active control still exists.
- The negative temperature coefficient of the samarium cobalt class of magnets could result in up to 60 percent strength increase, but this requires verification.
- Actuators based on the diamagnetic principle may now become a practical alternative.

Diamagnetic actuators utilize the property that diamagnetic materials are repelled by an external magnetic field. The main disadvantage of these actuators is the low load capacity per unit volume of material. The low load capacity, in the room ambient temperature range, is due to the relative permeability of the diamagnetic materials being only slightly less than one. Superconducting materials are a limiting case of diamagnetic materials, with relative permeabilities approaching zero. Now, with the development of materials that are superconducting above the temperature of liquid nitrogen (77°K), one element of the magnets-in-repulsion actuator may be replaced with a superconductor yielding another actuator alternative for low-temperature applications. Honeywell's finite-element modelling of the permanent magnets-in-repulsion actuator will probably be extended to include this actuator.

CONCLUSIONS

An overview of magnetic suspension actuators has been presented. Although certainly not complete, we have covered the main attributes of each actuator type and provided some insight into the trade-offs to be considered in the selection and design process.

We see the low-temperature and cryogenic applications area as one where magnetic suspensions incur additional advantages over mechanical suspensions, specifically because the mechanical isolation yields both partial thermal isolation and freedom from lubricant problems.

TABLE 1. - ACTUATOR COMPARISONS

Actuator Type	Electromagnet	Voice Coil	Variable Reluctance	P/M Repulsion
Active or Passive	Active	Active	Passive	Passive
Force Function	$F \propto I^2$ $F \propto I/G^2$	$F \propto I$	$F \propto$ Offset	$F \propto$ Offset
Cross-Axis Forces	Minimal	~ Negligible	Substantial	Substantial
Cross-Axis Torques	Significant	Minimal	Substantial	Substantial
Sensors Required	Postion and/or Flux	Position	None	None

TABLE 2. - ACTIVE ACTUATOR TRADE-OFFS FOR A SPECIFIC APPLICATION*

Actuator Type	Weight (kg)	Peak Power (watts)	Pk. Torque Error Due to Motions (N•m)	Hysteresis (N)
Electromagnet Low B and Hyst. Mat'l.	400	350	65	15
Electromagnet High B and Hyst. Mat'l.	125	225	50	160
Voice Coil Single Magnet (ID only)	400	850	130	--
Voice Coil Dual Magnets (ID and OD)	500	850	44	--

*Requirements
Force = 6500 N (1460 lbs)
Motion = ±2.5 mm Linear, ± 1 m Radian Rotation
Actuators located on 6.5 m diameter

TABLE 3. - COMPARISON OF THE FOUR ACTUATOR TYPES FOR SPECIFIC APPLICATION*

Actuator Type→ Parameter↓	Electromagnet	Voice Coil	Variable Reluctance	P/M Repulsion
Position Sensor	Yes	Yes	No	No
Radial Stiffness (K_R)	1×10^6 N/m(at 50μm) 10×10^6 N/m(at 5μm)	1×10^6 N/m(at 50μm) 10×10^6 N/m(at 5μm)	1×10^6 N/m	1×10^6 N/m
Radial Force	50 N (Full Range)	50 N (Full Range)	50 N (at 50μm)	50 N (at 50μm)
Axial Unbalance Stiffness (K_U)	~0	~0	0.5×10^6 N/m	1.5×10^6 N/m
Total Weight (per end)	0.6 kg	1.4 kg	0.5 kg	0.9 kg
Suspended Portion Weight	0.15 kg	1.3 kg	0.1 kg	0.4 kg
Peak Power	15 watts	50 watts	None	None
NOTES	Added Complexity But Higher Stiffness at Small Motions		Require Upsized Active Axial Support to Accommodate K_U	

*Assumed Application: Radial support of gimbal or wheel (per end)
Required radial stiffness (K_R) = 1×10^6 N/m (11.4×10^3 lb/in)
Radial motion range = ± 50 μm (± 0.002 in)
Minimum clearance = 200 μm (0.008 in)
Assumed position sensor resolution (where used) = 5 μm (0.0002 in)

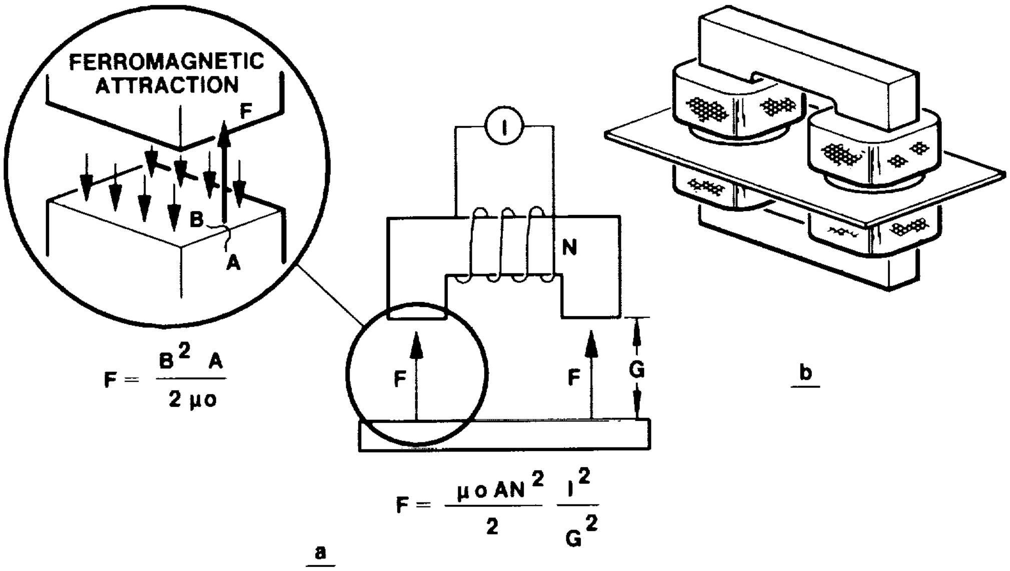


Figure 1. - Electromagnetic Attraction

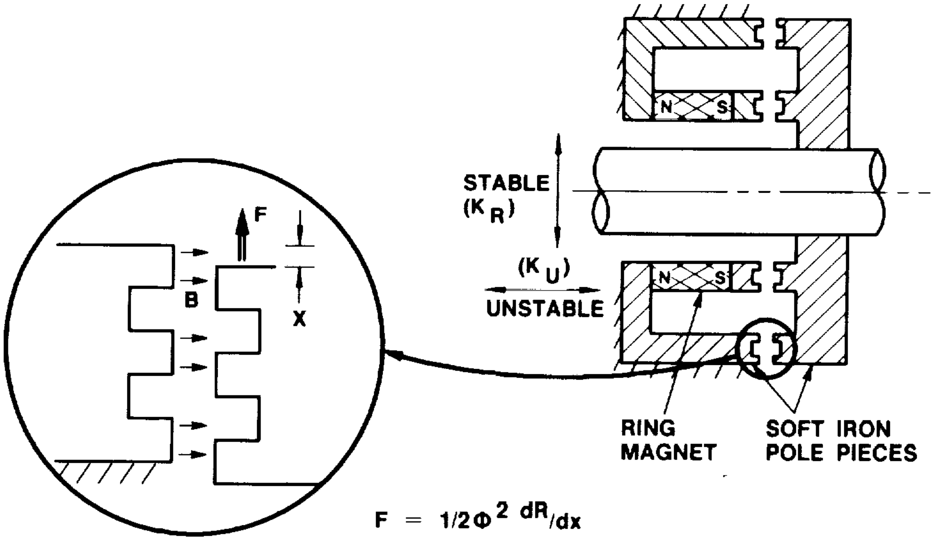


Figure 2. - Variable Reluctance Actuator

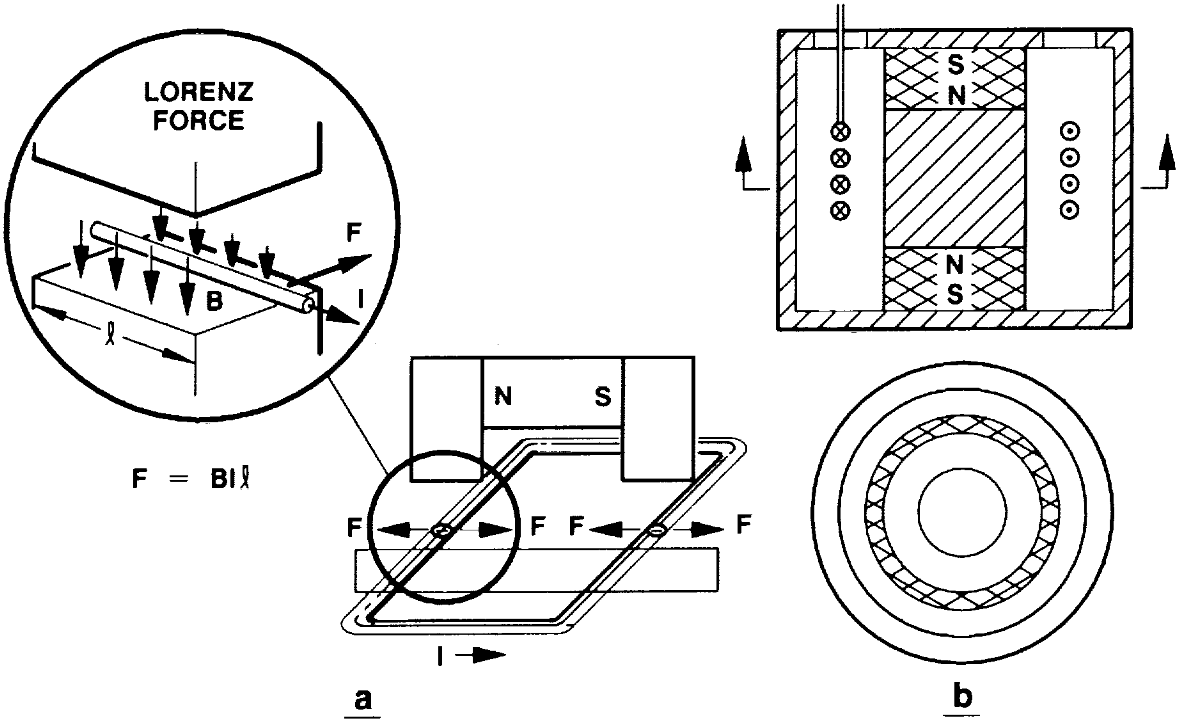


Figure 3. Voice Coil Actuator

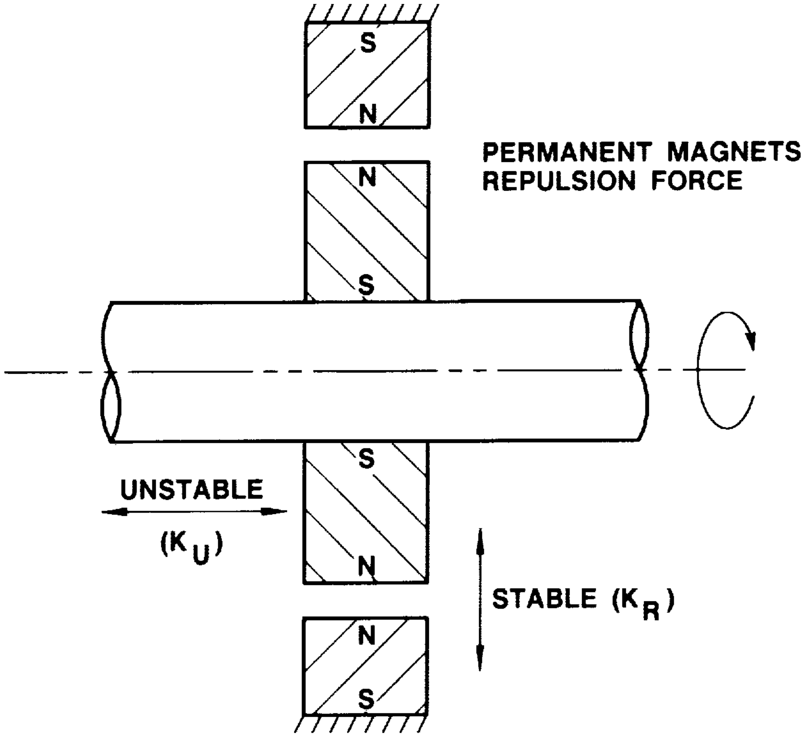
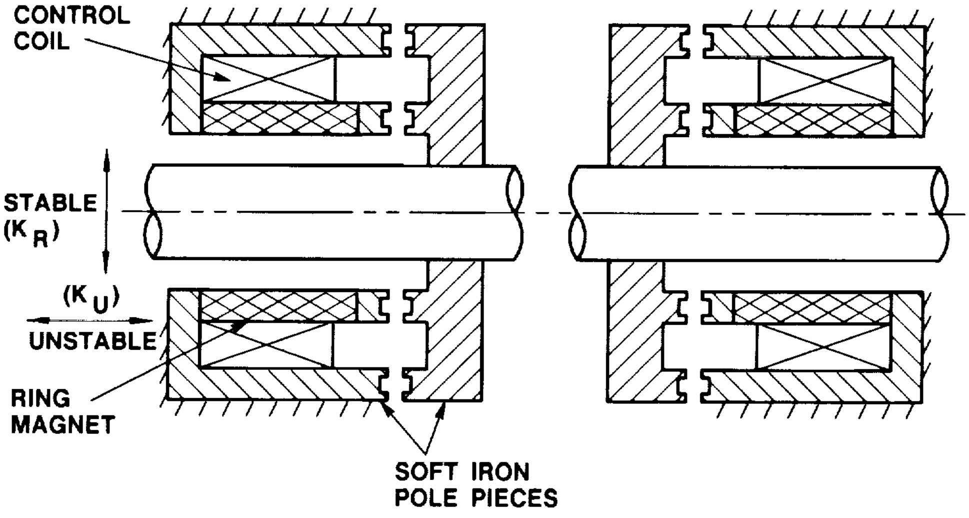
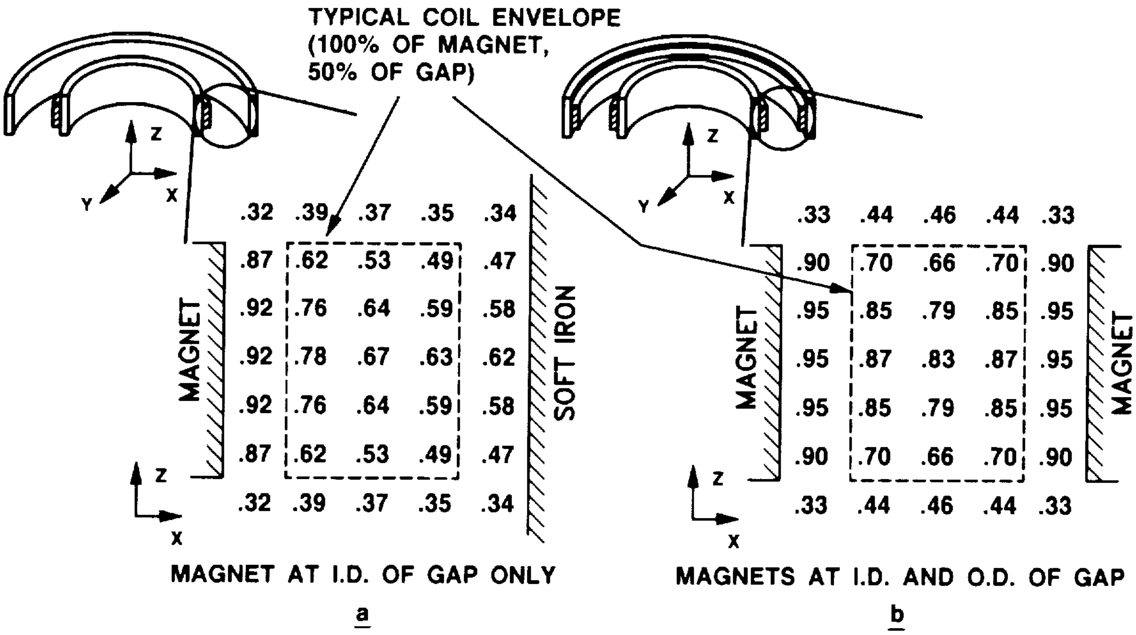


Figure 4. Permanent magnet (PM) Repulsion Actuator



VARIABLE RELUCTANCE, PASSIVE RADIAL SUSPENSION WITH INTEGRATED ELECTROMAGNET ACTUATOR FOR ACTIVE CONTROL OF THE UNSTABLE AXIAL AXIS (SIMPLIFIED CONFIGURATION)

Figure 5. Integrated Passive Radial, Active Axial Suspension



.32	.39	.37	.35	.34
.87	.62	.53	.49	.47
.92	.76	.64	.59	.58
.92	.78	.67	.63	.62
.92	.76	.64	.59	.58
.87	.62	.53	.49	.47
.32	.39	.37	.35	.34

MAGNET AT I.D. OF GAP ONLY

a

.33	.44	.46	.44	.33
.90	.70	.66	.70	.90
.95	.85	.79	.85	.95
.95	.87	.83	.87	.95
.95	.85	.79	.85	.95
.90	.70	.66	.70	.90
.33	.44	.46	.44	.33

MAGNETS AT I.D. AND O.D. OF GAP

b

Figure 6. Voice Coil Actuators - Magnetic Field Distribution