2nd International Symposium on Magnetic Beaing
July 12-14, 1990, Tokyo, Japan

MAGNETIC SUSPENSION FOR A TURBOMOLECULAR PUMP

G. GENTA*; L. MAZZOCCHETTI**; E. RAVA**

* Politecnico di Torino, c.so Duca degli Abruzzi 24, 10129 Torino, Italy
** Elettrorava S.p.A., Via Don Sapino 176, 10040 Savonera - Torino, Italy

Abstract

Magnetic bearings allow to solve in a very effective way many of the problems encountered in high vacuum turbomolecular pumps: lubrication under vacuum, hydrocarbon contamination, operating position, vibration and noise.

The development of a magnetically suspended turbomolecular pump based on a "one active axis" configuration is here described. A permanent magnet radial passive bearing and one active axial bearing with its feed-back control loop have been developed and thoroughly tested.

Before building the prototype, a hybrid turbomolecular pump which uses the same radial bearing and one axial ball bearing has been built and tested. As the performance of this machine proved to be very satisfactory, the prototype of the machine with full magnetic suspension is now under construction.

1. Introduction

Rotors operating in high vacuum and at high speed are a very interesting application field for magnetic bearings. The substitution of rolling elements bearings with magnetic suspension systems in turbomolecular pumps, in which both these aspects are present, is very promising.

A turbomolecular pump is a high vacuum device, designed to operate in the range between 1 and 10^{-8} Pa, therefore needing a backing pump, which is normally of the vane type. The rotor is made of a series of bladed discs which look like axial compressor discs but work in a completely different way as they operate in a free molecular flow with the well known principle of molecular pumps [1].

Turbomolecular pumps are powered usually by asyncronous high frequency a.c. motors or d.c. brushless motors located in the vacuum container; types driven by belts through magnetic couplings were also produced.

The spin speed of these machines can range from 10000 up to 90000 rpm depending on the size of the pump, and the bearings must be located within the vacuum container in order to avoid high speed seals on the shaft. It is true that the bearings can be located on the side in which the pressure is higher (output side of the pump) but the pressure is at any rate of about 1 Pa and consequently very low vapour pressure lubricants are needed.

Standard turbomolecular pumps with high speed ball bearings are commercially available since many years, but they cannot completely satisfy the most demanding applications. Although high speed ball bearings can operate under vacuum, there are still some problems to be solved in a completely satisfactory way, mainly due to the presence of lubricants, which can cause vacuum contamination by hydrocarbons, the need for high speed balancing of the rotors and the limited service life.

Residual vibration and noise can also be a problem in some special applications and when the pump is used in a laboratory environment close to people.

All these problems can be avoided with the use of magnetic bearings, which are increasingly used in order to achieve higher speed, cleaner vacuum, lower vibration and noise levels and better reliability.

Hybrid configurations, i.e. configurations employing a combination of magnetic and mechanical bearings, have been used for a few years, but all manufacturers are now involved in the development and sometimes already in the regular production of turbomolecular pumps with total magnetic suspension.

Most of these more advanced solutions are based on five active axes magnetic suspensions which have been developed since some years and run properly, but their cost is still too high and, as a consequence, these pumps have not reached a wide diffusion on the market.

The tendency is now to build turbomolecular pumps with total magnetic suspension reducing to a minimum the number of the actively controlled axes. According to Earnshaw theorem, one of the possible solutions is to have four radial passive axes and only one axial active axis.

This means that permanent magnets or uncontrolled electromagnets are used for the radial bearings, whereas the axial position of the rotor is controlled by an active magnetic bearing.

Another advantage of the total magnetic suspension is that the sensitivity to unbalance is much lower because the rotor tends to rotate around its principal axis of inertia. Therefore the balancing requirements are far less strict than in conventional solutions and in most cases it is not necessary any more to balance the rotor at its operating speed.

The less strict balancing requirements lead to a potential reduction of production costs which can, at least in part, compensate for the higher costs of the suspension system.

Owing to the above mentioned reasons, a new turbomolecular pump with magnetic suspensions is now being developed according to the scheme of fig.1.

The rotor is suspended by means of two passive radial magnetic bearings and one active axial magnetic bearing. This

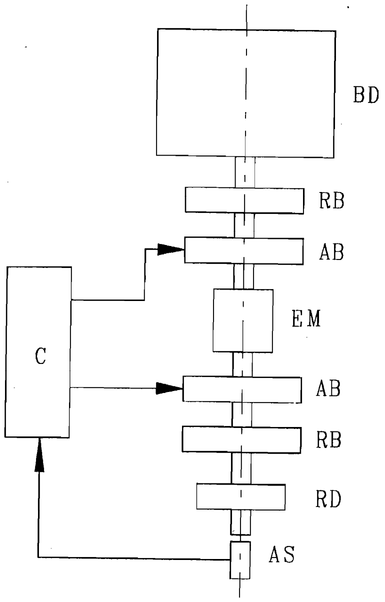

Fig. 1. Scheme of the magnetic suspension system. AB: active axial magnetic bearing; AS: axial position sensor; BD: bladed discs; C: electronic control unit; EM: electric motor; RB: passive radial magnetic bearing; RD: passive radial magnetic damper.

solution is well suitable for a turbomolecular pump, where the radial loads are negligible and the required radial stiffness is within the capabilities of permanent magnets.

Therefore only the axial position has to be controlled actively in order to compensate for the negative stiffness of the passive radial magnetic bearing which are inherently axially unstable. The disadvantage of this configuration is that a passive radial damper is needed in order to be able to accelerate beyond the critical speeds during startup.

A particular type of passive radial magnetic bearing has been developed for this particular application in order to obtain a very simple and low-cost system. In this development stage several bearings of all types shown in fig. 2 have been built and tested. A small turbomolecular pump with hybrid suspension system with one passive radial magnetic bearing of the type considered and one traditional grease lubricated ball bearing has been built and thoroughly tested.

2. Passive radial magnetic bearings

2.1. Bearing type A In order to find the most suitable passive radial magnetic bearing for this application, different types have been examined comparing the following parameters: radial stiffness, axial instability, ease of manufacturing, ease of assembling and cost.

The radial and axial stiffness have been measured experimentally for the four main types shown in fig. 2.

The first type, shown in fig. 2a, is built-up of two concentric sets of annular permanent magnets axially magnetized. The magnets of each set are separated by thin aluminium rings and repel each other axially, forcing the flux lines of each magnet to close on themselves in the radial air gap between the two sets of magnets.

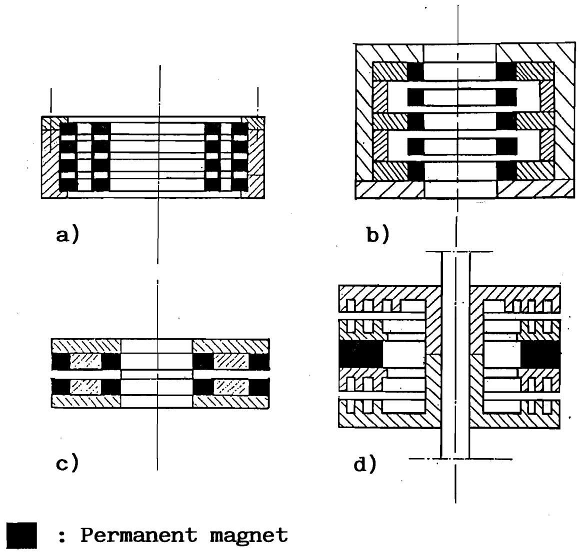

Fig. 2. Configurations of the passive radial magnetic bearing considered for the turbomolecular pump.

The two concentric sets of magnets repel each other radially. The radial stiffness is proportional to the number of magnets that have been employed and of course depends on the material of the magnets. Due to the very high demagnetizing force acting on the magnets, Samarium Cobalt (Sm-Co) or Neodymium Iron Boron (Nd-Fe-B) alloys have to be used for this kind of bearing.

2.2. Bearing type B A number of equal axially magnetized annular magnets attract each other and are alternatively rotating and fixed [2], [3]. At least two magnets are needed, but normally an odd number (5, 7, 9) is used in order to have a vanishingly small axial force when the rotating magnets are axially centred with respect to the fixed ones.

If the rotating magnets are shifted radially, a magnetic radial shear force that tends to restore the central position results. Furthermore, an external iron magnetic path can be built in order to close the flux and maximize the radial stiffness, as shown in fig. 2b.

2.3. Bearing type C Another type of passive radial magnetic bearing is shown in fig. 2c. The fixed and the rotating parts have the same geometry and attract each other. Each part is built up of two concentric axially magnetized annular magnets with an iron disc to close the magnetic flux. The radial stiffness is here again due to radial magnetic shear forces, but the axial force is not zero and increases with decreasing air gap.

2.4. Bearing type D The three types of passive radial magnetic bearings that have been described above have good radial stiffness and have already been employed in turbomolecular pumps for many years, but they have some disadvantages. In all types there are rotating magnets, which makes the construction more complicated and more costly.

The mechanical strength of Sm-Co and Nd-Fe-B materials used for the magnets is not high enough to withstand the high rotational speeds required in turbomolecular pumps if they are not prestressed by a surrounding aluminium ring. This ring has to be connected to the magnet by adhesive bonding or shrink fitting but in the latter case the thermal cycles could change the magnetic properties of the permanent magnets. Other construction types involving a welded magnet housing are also used.

Furthermore each bearing is built up of many small magnets which are brittle and have to be handled with care, so that the assembly in mass production is critical.

In order to avoid these problems, the passive radial magnetic bearing type D shown in fig. 2d has been developed for the magnetically suspended turbomolecular pump. The main advantage is that there is only one relatively big magnet which is not rotating, and the magnetic flux is guided through the air gap by some small iron annular teeth of suitable shape creating the radial stiffness.

The steel rotor shaft is then used to close the flux. When there are equal air gaps on both sides of the magnet, the axial force acting on the rotor is zero, but this is an unstable equilibrium.

Experimental measurements have been carried out on a passive radial magnetic bearing of the type D with a Sm-Co magnet with outer diameter of 56 mm, inner diameter of 34 mm and axial thickness of 10 mm. The results are shown in fig. 3, where the axial and radial forces are plotted as functions of the axial and radial displacements respectively.

A radial stiffness of about 100 N/mm was found with an air gap of 0.5 mm. The maximum axial force with zero air gap on one side and 1 mm air gap on the other side was 150 N.

3. Hybrid turbomolecular pump

The passive radial magnetic bearing type D has been at first used in a 60 l/s turbomolecular pump running at 72000 rpm. A cross section of the pump is shown in fig. 4.

There is one grease lubricated ball bearing and one passive radial magnetic bearing type D located near the centre of gravity of the rotor. The ball bearing is located at the lower end of the shaft and sustains the axial load. The surface of the iron teeth has to be machined with a spherical shape with the centre in the ball bearing.

The bladed discs are above the passive radial magnetic bearing so that there is no permanent magnet exposed to high vacuum or to other dangerous environmental conditions which can exist inside the vacuum chamber, as high temperatures. The damping of the system is quite low, but the rotor runs properly at 1200 Hz (72000 rpm), which is a much higher frequency than the first critical speed of the system which is at about 46 Hz (2760 rpm).

Quite a large amplitude of the synchronous whirl is obtained when crossing the critical speed and an amplitude limiter consisting of a ring of a special low friction material located around the shaft is used.

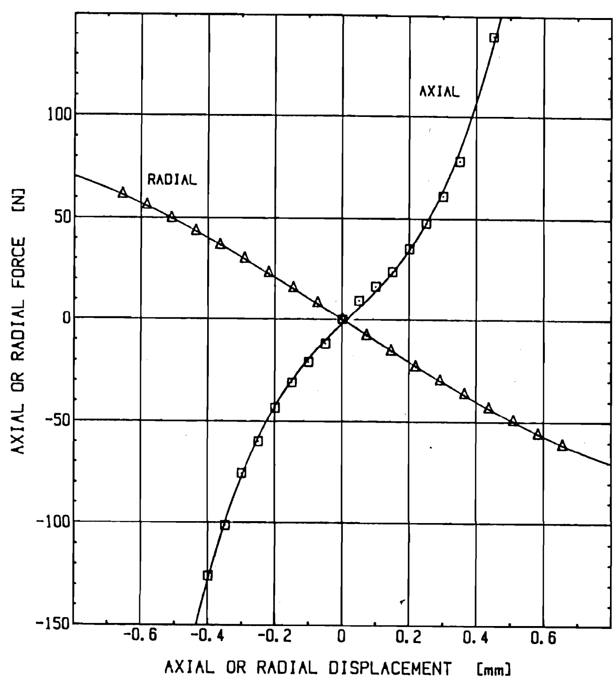

Fig. 3. Experimental characteristics of a passive radial magnetic bearing of type D.

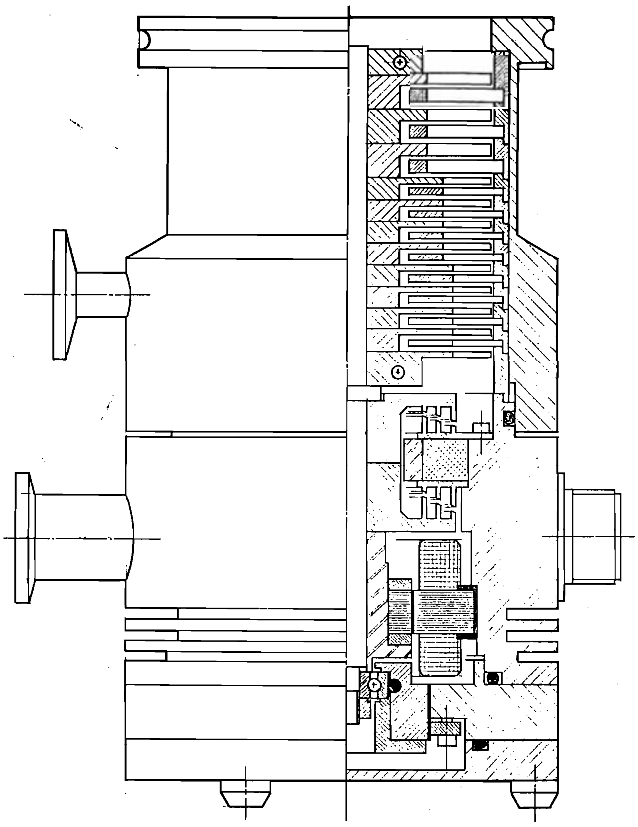

Fig. 4. Cross section of the hybrid turbomolecular pump.

The axial position of the rotor can be adjusted during assembly in order to let the passive radial magnetic bearing work around the zero axial force position. The a.c. squirrel cage motor has a total power consumption of about 30 W under vacuum.

The rotor has been balanced at 1500 RPM on a balancing machine and higher speed balancing was not needed.

The dynamic behaviour of the rotor was investigated using DYNROT finite element code developed at the Mechanics Dept. of Politecnico di Torino [4]. The radial force-displacement characteristic of the magnetic bearing obtained experimentally was linearized about the equilibrium position. A total of 22 nodes and 21 Timoshenko beam elements has been used.

The critical speeds, mode shapes and the Campbell diagram shown in fig. 5 were computed.

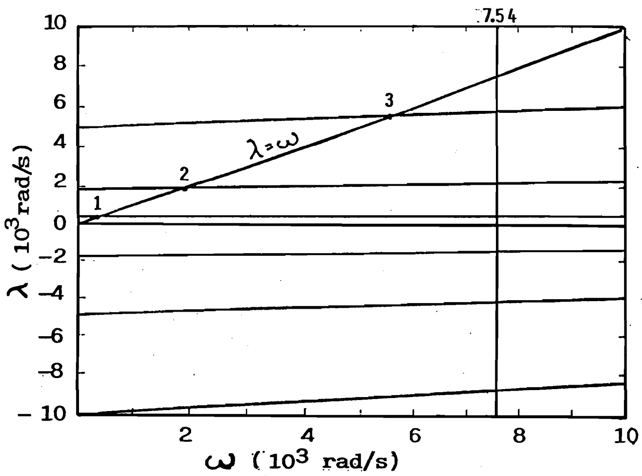

Fig. 5. Campbell diagram of the hybrid turbomolecular pump.

The first critical speed computed by the finite element code is 2730 rpm. Other critical speeds within the working range were obtained, as seen from the Campbell diagram. The analysis of the relevant mode shapes shows however that they are little excited by the static unbalance of the bladed discs and heavily damped by the lower bearing.

The numerical results were later confirmed by vibration measurements on the prototype.

Strong vibrations were detected at a speed of 2760 rpm, with very good accordance with numerical critical speed predictions. As expected, no other critical speed was detected up to operating speed.

The vibration spectrum at 72000 rpm is shown in fig. 6. The four peaks are due to synchronous whirl (1200 Hz), disturbances due to the ball bearing (448 Hz), whirling in the first forward mode (48 Hz) and disturbances coming from the vane backing pump (24 Hz).

The pump was thoroughly tested in operating conditions for some hundred hours with very satisfactory results. In particular the very low energy consumption, about 1/2 of that of conventional solutions and the very low level of vibration without the need of fine balancing were the most interesting features of the machine.

4. Turbomolecular pump with completely magnetic suspension

4.1. General description Owing to the success obtained with the hybrid machine, the design of the pump with a completely magnetic suspension was undertaken.

A cross-section of this machine, a 360 l/s pump running at 48000 rpm, is shown in fig. 7.

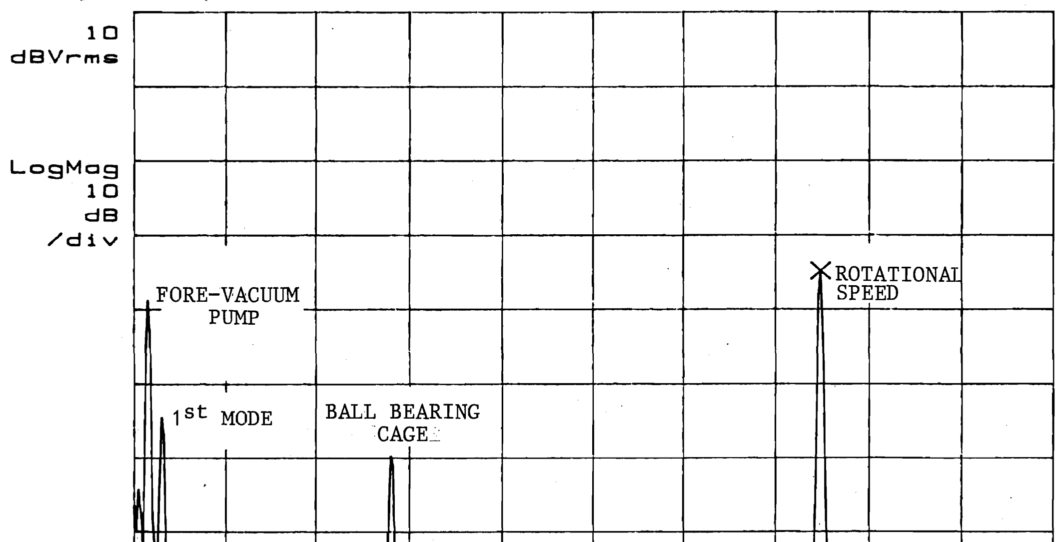

Fig. 6. Vibration spectrum at 72000 rpm.

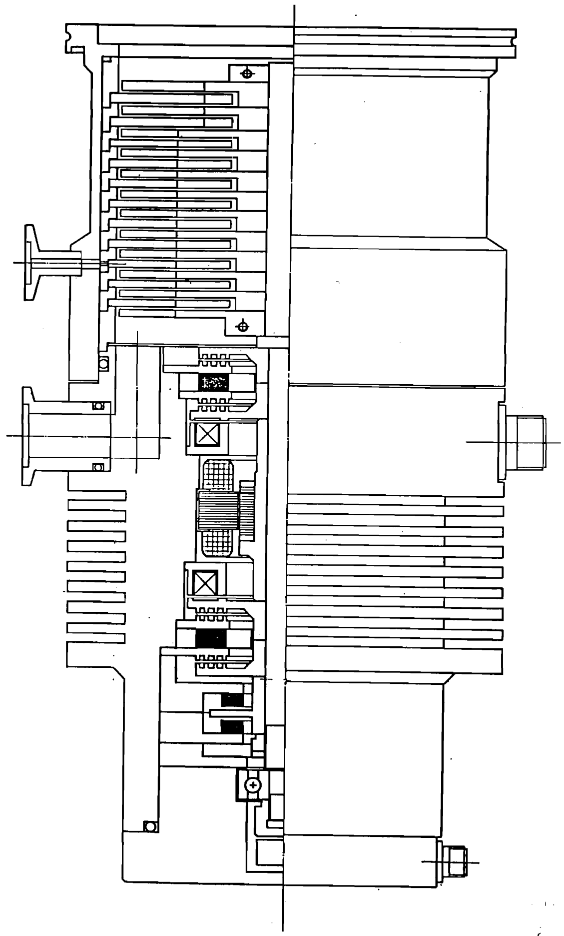

Fig. 7. Cross section of the turbomolecular pump with full magnetic suspension.

Two passive radial magnetic bearings of type D are employed; the upper bearing is located close to the centre of gravity of the rotor. The axial actuator is split into two electromagnets, each one attracting one iron disc of the passive radial magnetic bearing, so that the rotor can be attracted in opposite directions.

The five axes magnetic suspension is completely symmetrical with respect to the a.c. squirrel cage motor. Below the second passive radial magnetic bearing there is an eddy current radial damper [2], [5] consisting of an aluminium disc rotating between two axially magnetized annular Sm-Co permanent magnets. The two magnets attract each other and the flux lines are crossing the conductive disc, whereas an external iron path is closing the flux.

If the disc does not move radially, there is no magnetic flux variation and no current in the disc. But if there is a radial movement of the disc due to whirl motion, eddy currents are induced and the radial motion is damped. At the lower end of the shaft there is a dry-lubricated touch-down ball bearing and the axial inductive sensor to provide the axial position signal to the electronic control system.

It can be seen from the previous description that there is no rotating magnet in the system, which is an advantage from several points of view. Between the magnet of each passive radial magnetic bearing and the rotating shaft there is a ring of a special anti-friction material in order to limit the radial displacement of the rotor if the damper is not sufficient in case of an external shock.

<u>4.2. Active axial bearing</u> The electromagnet used for the axial magnetic bearing is shown in fig. 8a. All relevant data are reported in tab. 1.

If the reluctance of the iron path is neglected the axial force is given by

$$F = \frac{\mu_0 S_g N^2 I^2}{4d^2} \tag{1}$$

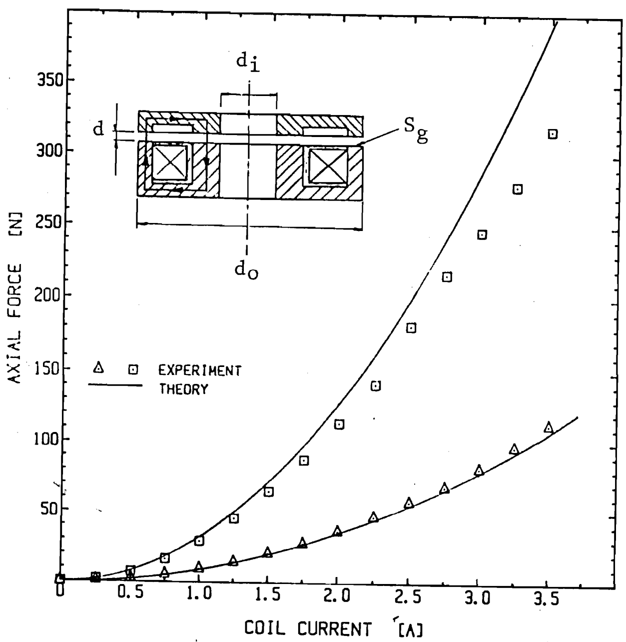

Fig. 8. Axial active magnetic bearing. a) Axial electromagnet. b) Force-current characteristics of the electromagnet.

Outer diameter d_o	80 mm
Inner diameter d_i	20 mm
Air gap surface S_g	1200 mm^2
Number of turns N	160
Wire diameter d_w	1 mm
Nominal air gap d	0.5 mm
Coil resistance R	0.64 Ω
Coil inductance L	35 mH
Nominal load F	410 N
Nominal current density j	4.5 A/mm^2

Tab.1. Data of the axial bearing

It can be seen that the force is proportional to the surface of the air gap, to the square of the current, whereas it decreases with the square of the air gap d.

A plot of the force as a function of the current for the electromagnet of fig. 8a is shown in fig. 8b. The theoretical curve and the experimental points are both plotted and the agreement is quite satisfactory.

The two passive radial magnetic bearing are axially unstable: when the rotor is centred the axial force is zero, but a small deviation from the central position will cause the rotor to move rapidly out of this position. Therefore the axial magnetic bearing must be active and an electronic control unit providing the required stability is needed.

The situation is graphically illustrated in fig. 9, where the force F_{mz} due to the passive radial magnetic bearing acting on the rotor is shown as a function of the axial co-ordinate z of the rotor. As a first approximation in practice it can be assumed that the force F_{mz} increases linearly with z

$$F_{mz} = -K_{mz} z \qquad (K_{mz} < 0) \tag{2}$$

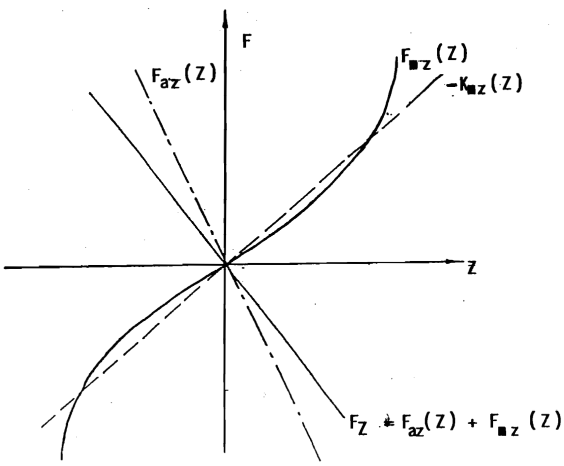

Fig. 9. Force-displacement curves for axial stabilization.

The constant K_{mz} is the negative axial stiffness of the passive radial magnetic bearing. In order to make the system stable, a force

$$F_{az} = -K_{az} \cdot z \qquad (K_{az} > 0) \quad , \tag{5}$$

generated by the electromagnets driven by the control system that monitors continuously the position of the rotor, must be applied. Constant K_{az} is positive and must overcome the intrinsic axial instability of the passive bearing.

The resulting axial force acting on the rotor is given by

$$F_z = F_{mz} + F_{az} = -K_z \cdot z \quad , \tag{4}$$

where constant K_z is the overall axial stiffness of the system and has to be positive in order to achieve stability. The condition for stability is then

$$K_z > 0 \quad , \quad \text{i.e. } K_{az} > | K_{mz} | \quad . \tag{5}$$

As it is shown in fig. 9, the slope of F_{az} has to be greater in absolute value then the slope of F_{mz}. Of course this condition allows to achieve just static stability and other conditions has to be fulfilled by the control system in order to get dynamic stability.

4.3. <u>Control system</u> The block diagram of the control system is shown in fig. 10. The position monitoring is performed by an inductive sensor giving an electric signal which is proportional to the distance between the sensor and the end of the shaft.

The signal is compared with a stable reference signal in order to get a signal proportional to the position z of the rotor. It is then amplified and differentiated to get the information about the axial velocity of the rotor in order to achieve the required damping. The position and velocity signals are then added together to drive the final amplifiers.

The latter are voltage driven current amplifiers that inject the proper current into the coils of the electromagnets. The two electromagnets are working alternatively to attract the rotor towards the central position.

Following a displacement of the rotor in the positive direction z, the current in coil 1 is zero whereas the current in the other coil increases to generate a restoring force attracting the rotor back to its central position.

If otherwise the displacement occurs in the negative direction -z, the current in coil 2 is zero whereas the current in coil 1 increases. Therefore the signal has to be divided in two paths and rectified before driving the final amplifiers. Furthermore another signal must be added to get the stability around the equilibrium point.

The axial force of each electromagnet increases quadratically with the current, and if the current in one coil starts increasing exactly when the current in the other coil approaches zero, the force-displacement curve has a zero derivative in the origin.

This condition is shown in fig. 11, where it can be clearly seen that a negative stiffness is obtained around the origin when the force of the active axial magnetic bearing is added to the axial force of the passive radial magnetic bearing.

This problem can be solved if a current I_o is injected in both coils also when the central position is obtained (z=0), which can be achieved by adding a constant voltage to the signals driving the two power amplifiers, as shown in fig. 10.

4.4. <u>State of development of the project.</u> At present the construction of the prototype has not yet been completed.

Numerical simulations and particularly rotor dynamics computations have been performed using the already mentioned DYNROT code in order to verify the ability of the system to run correctly.

A prototype of the active magnetic axial bearing complete of its control system has been built and thoroughly tested. The conditions for both static and dynamic stability have been achieved.

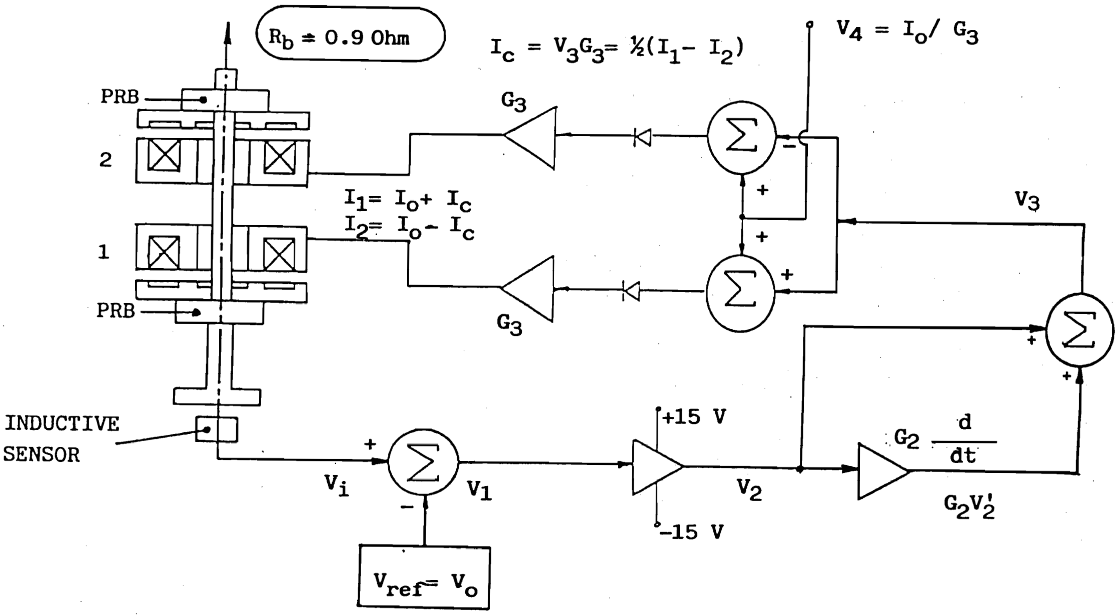

Fig. 10. Block diagram of the control system.

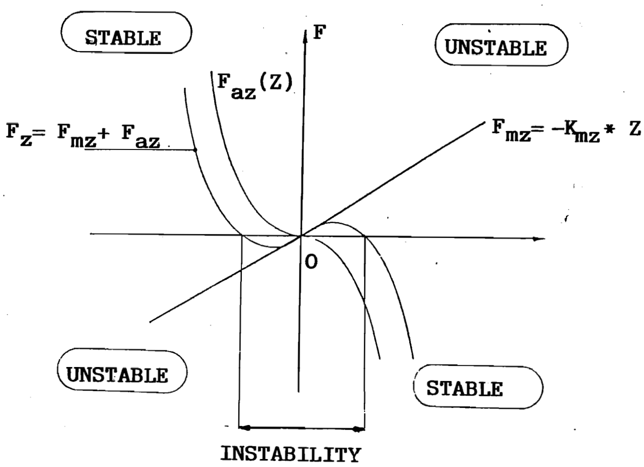

Fig. 11. Instability about the zero-force position.

As all components have been tested separately and some of them have operated in a satisfactory way on the hybrid machine, the turbomolecular pump with full magnetic suspension is expected to fulfil its design goals.

5. Conclusions

The passive radial magnetic bearing of the type shown in fig. 2d seems to be an attractive device for many applications where the manufacturing cost and the ease of assembly are the determining factor in the choice of the magnetic bearing.

Furthermore the absence of rotating magnets makes it very suitable for high rotational speeds. Using this kind of passive radial magnetic bearing a simple total magnetic suspension for turbomolecular pump can be built with only one actively controlled axis.

This leads to a simple control system that has to control only the axial position of the rotor. The main drawback of this type of suspension is the lack of damping of the passive radial magnetic bearing which can create some problems especially during acceleration and deceleration of the rotor. An eddy current damper has been added, but further development can be needed to get better solutions.

In order to verify the suitability of this radial magnetic bearing, a hybrid turbomolecular pump was built and tested.

The hybrid machine fulfilled all design goals, proving to be able to reach the design operating speed and to work satisfactorily for prolonged periods of time.

In particular it showed a very low power consumption and the possibility of operating with very low noise and vibration levels without the need of fine balancing. The use of ball bearings with ceramic rolling elements is now considered in order to improve the performance of hybrid turbomolecular pumps.

The axial active bearing has been tested separately and the final prototype is under construction.

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