CONSIDERATION FOR THE DESIGN OF THE DRIVING AND BEARING SYSTEM FOR LARGE TELESCOPES

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1. INTRODUCTION

The Astronomers target for the present and the next decade is the construction of large telescopes 3,5 - 4 m. primary mirror diameter and very large telescopes 8 - 9 m. primary mirror diameter, in the array or multi-mirror design, improving at the same time the performance of the instruments compared to the efficiency and precision of the same.
Within the limits of the feasibility study for the 3,5 m. italian national telescope, the writhers have investigated the alternative possible design for the supporting and driving system for the AZ and EL axes in order to verify the possibility and ability to obtain the requested performances

- absolute pointing with pointing model, less or equal to 2 arcsec RMS
- drift tracking errors less than or equal to 0,3 arcsec in 30 min.
- max. angular speed of both axis, 1^{o}/sec
- acceleration and deceleration time, 1,5 sec
- max. tracking acceleration, $0{,}03^{o}/sec^{2}$
- max. torque caused by wind gusts on the elevation axis, 6000 Nm
- max. torque caused by not equilibrated masses on the elevation axis - 2000 Nm

The telescope is an alt-azimuth mount having the advantage of gravity caused deformations symmetric and linear-function of the EL axys, thus easyly modellable.

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The expected performances and the demanded working tolerances has suggested to the designers to investigate various solutions for the support and guide systems, making finally a comparison both from the cost and the precision of response point of view.

The solutions and comparisons herebelow illustrated can be considered also for very large telescopes.

2. BEARING SYSTEMS

The type and the construction accuracy of the supporting system are relevant factors in obtaining the requested performances, because they influence directly the geometry of the axys and, through their friction torque, in a very important way the smoothness of the regulation of the driving system.

Three alternative solutions have been studied.

- Hydrostatic-pads supporting system (see fig.1)
- Ball bearing supports
- Roller bearing supports

Solutions that have been compared on the basis of the following performance-characteristics

a) Accuracy of rotation

This characteristic is sufficiently good for all studied solutions, with a slight advantage for the hydrostatic pads.
In fact the ball and roller bearings have a small indefinite component on the rotation accuracy, caused by the small dimensional differences if the revolving elements, defect missing in the hydrostatic system that has an error in rotation exactly repeatable at every revolution with very slow fluctuations.

b) Friction torque

The friction torque has a typical diagram that distinguishes the revolving supports from the hydrostatic ones (see fig.2)

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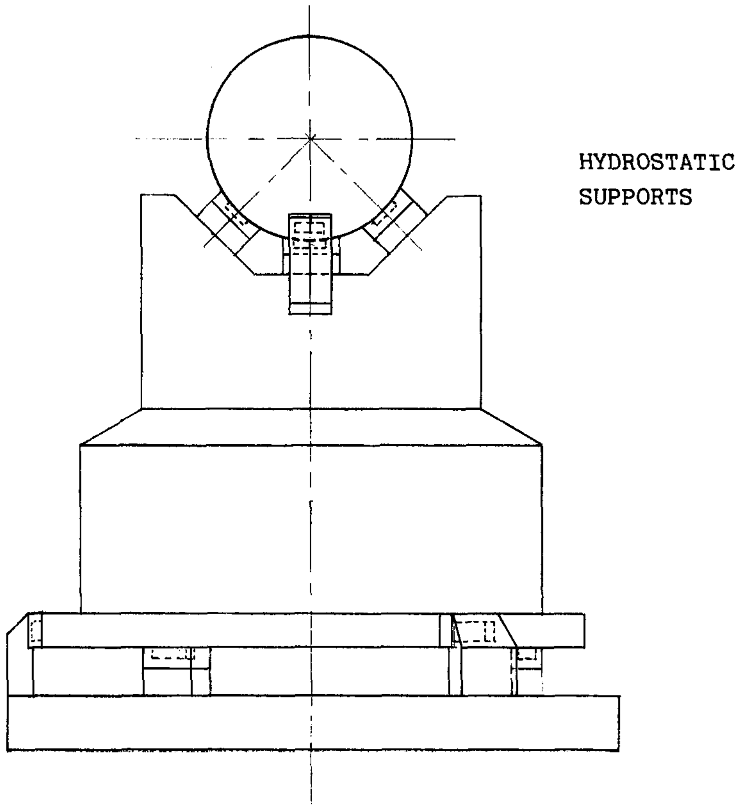

fig.1

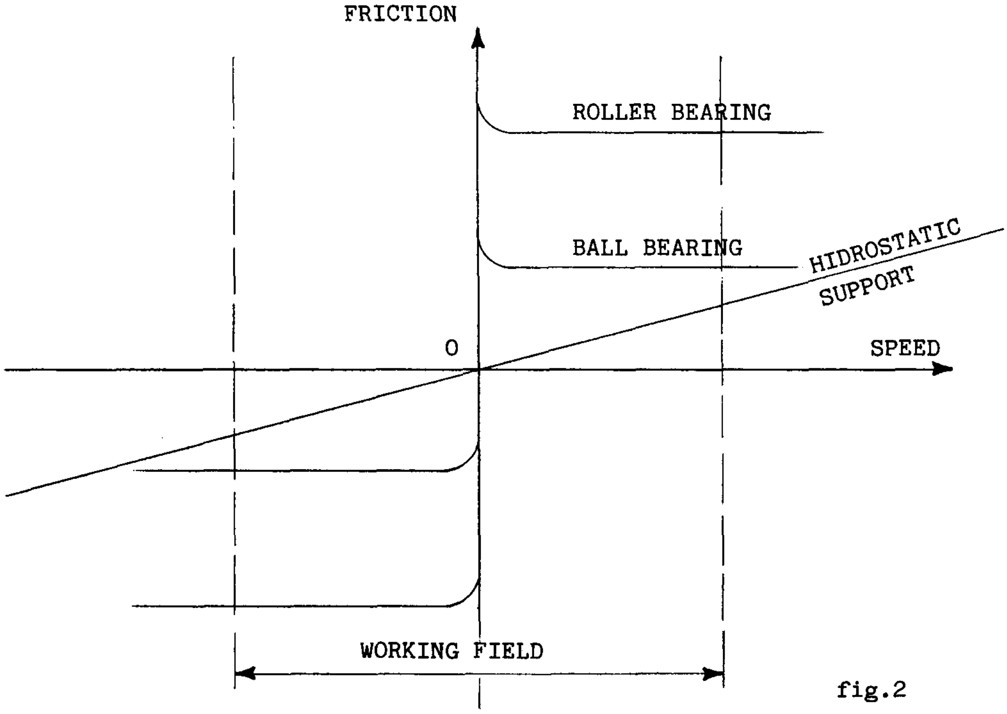

fig.2

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Both the ball and roller bearings have a friction torque nearly constant at every speed, except a light encrease near the zero speed.

This reduction of the friction coefficient with the speed can cause the "stik-slip" phenomenon at the very slow speeds, revealed by small skippings of the axis; the trouble becomes greater, the higher the elasticity of the driving system is.

The hydrostatic support systems, on the contrary, have a friction torque of the viscous type, that is directly proportional to the speed, having moreover a dampening effect that improves the dinamyc performances of the axis.

At the tracking speed the viscous friction torque is very small if compared with the one of the ball and roller bearing, and consequently the pointing accuracy is proportionally encreased.

Herebelow are listed the friction torque figures for the AZ axis in the three examined solutions:

- hydrostatic supports	100 ÷ 150 Nm
- ball-bearing	2500 ÷ 3000 Nm
- roller-bearing	10000 ÷ 11000 Nm

c) Rigidity

The bearings rigidity figure gives a weighty contribution to the ability of the system to maintain the predetermined position and geometry against non-controllable external disturbing torques.

Moreover, the bearings rigidity influences directly the torsional resonance frequency of the supported axis, and consequently the performances that can be obtained in the positioning control system.

The calculated figures for the AZ axis are:

- hydrostatic supports	54.000 N/μm
- roller-bearings	43.000 N/μm
- ball-bearings	8.300 N/μm

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d) Temperature

The bearings temperature must be as close as possible to the one of the telescope structure in order to avoid any turbolence of the air, with consequent seeing worseming, caused by convection and radiation.

The ball and roller bearings do not cause valuable heat production and consequently have no influence on the seeing quality.

In the hydrostatic supports must be taken in account a temperature encrease of 5 ÷ 6 ^{o}C caused by the drawing of the oil in the pads and the laminar resistors.

To reduce the problem in negligeable dimensions, it has been decided to introduce in the hydraulic circuit a regulated cooling unit so that the oil reaches the pads at a temperature lower than the ambient one, contacting the telescope structure pratically at the same ambient and structure temperature (see fig.3)

e) Reliability and availability

From this point of view the hydrostatic solution is the most interesting, because of the absence of any wear of the guides and the easiness of maintenance of the hydraulic circuit components.

Also with the ball and roller the maintenance problems are negligeable, but the possibility of any breaking must be faced with the change of the complete bearing, with costs and delivery time very important.

CONCLUSIONS

Having compared the three different systems on the base of quality of performances and from the costs point of view, the suggested solution has been the hydrostatic support for the AZ axis, whilst for the EL axis the choice between hydrostatic or preloaded roller bearing is open to the influence of other factors like type of motors and optical instruments dimensions.

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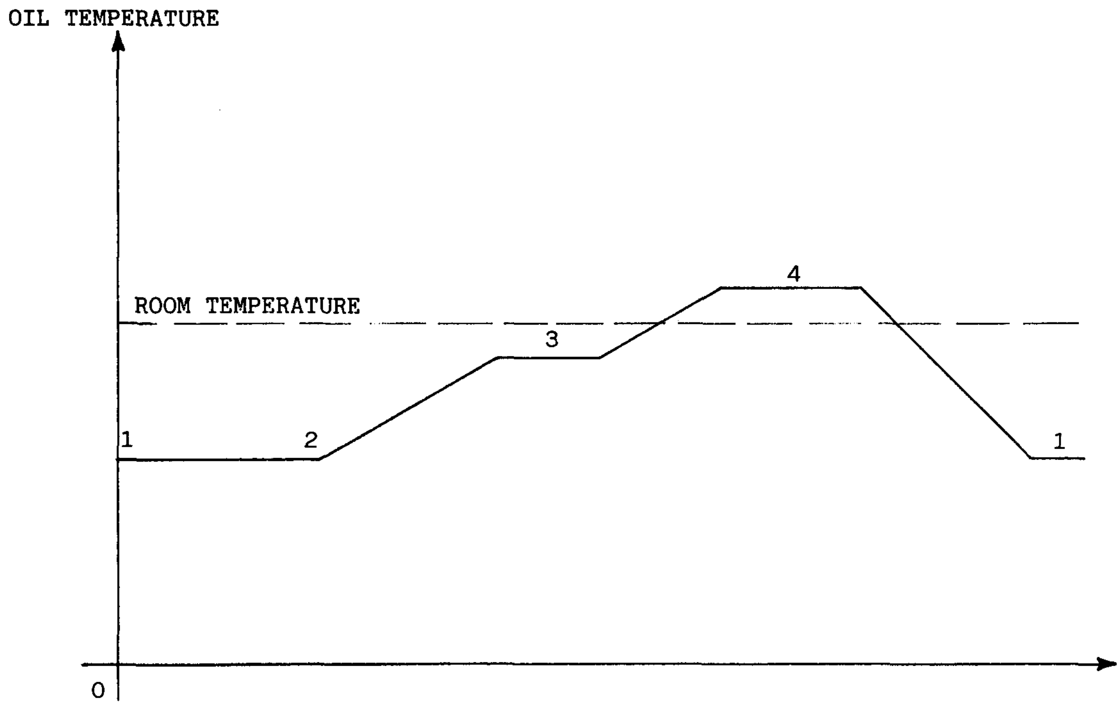

fig.3

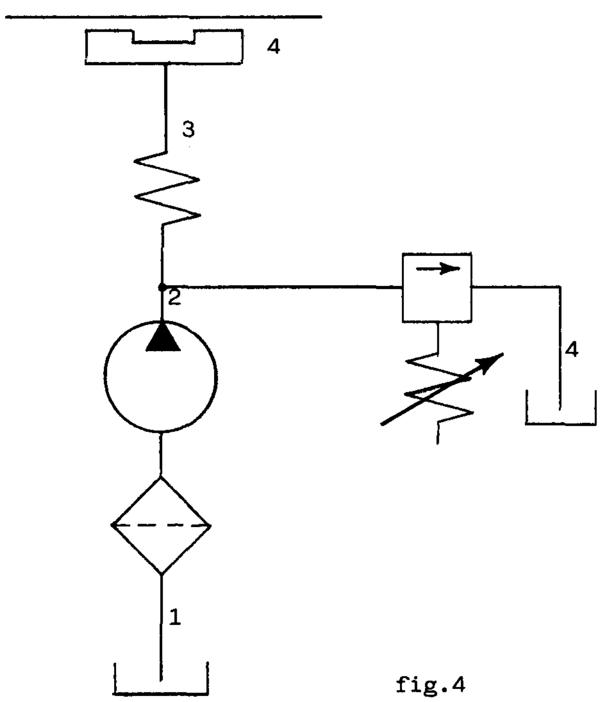

fig.4

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It must be enphasized that on the EL axis the use of roller bearings determines a friction torque that is far less important than the one on AZ axis, and consequentely other factors like dimensional and thermal aspects determine the choice.

The hydrostatic supports are foreseen with a constant pressure feeding and laminar resistors, in alternative at the solution with indipendent pumps, both for economic and reliability reasons, as the difference in rigidity is negligeable (see fig.4)

3. <u>DRIVING AND ENCODING SYSTEMS</u>

The driving and encoding systems are relevant in obtaining the telescope requested performances, and they must grant an high precision and quick response along with an elevated resolution and precision in the measuring system.

The quick response, and consequently an elevated bandwidth, are requested to reduce the temporary errors due to wind gusts, that cause disturbing torques on the EL axis with relevant components at high frequencies.

The target of the study was the obtaining a bandwidth of the space loop of both axis of 3 HZ min., achievable with a torsional resonance frequency of 9 HZ min. of both axis.

To the driving systems have been thus requested high rigidity toghether with lack of blacklash and low reverse friction torque.

The solutions that have been investigated are the following:
- two motors and two preloaded gear and pinion
- two motors and two preloaded friction wheels
- torque motor on axis

For all investigated solutions have been evaluated the different quality factors, i.e.:

a) <u>Reverse blacklash</u>

Do not exist for all the three solutions

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b) Rigidity

It is very high in the solution with torque motor on axis, because is missing any mechanical transmission between motor and driven axis.

For the other two solutions, it is feasible, without big difficulties, a transmission with rigidity of $1,6.10^9$ Nm/rad min., figure sufficient to obtain on both axis the demanded torsional resonance frequency.

Any case, the type of preload torque that must be used for the two investigated solutions is the differential type, (see fig.7) in order to obtain a global rigidity proportional to the parallel of the two cynematic chains.

When using the two friction wheels instead of the two preloaded gear and pinion, with the same dimensions of the reducers, we can obtain a rigidity figure slightly lower owing to the pins in the gear boxes structure (see fig.5).

c) Reverse friction torque

The reverse friction torque is null in the torque-motor on axis solution, and presents the max value in the friction wheel drive solution owing to the high revolving torque of the loaded wheels.

In the solution with two preloaded gear and pinion, the reverse friction torque on both axis has satisfactorily small figure, specially if is employed an adaptiv preload, i.e. a preload value variable in direct proportion with the disturbing external torque (see fig.6 - fig.7).

Of course the value of the time constant of encreasing and decreasing of the preload torque must be carefully determined, very small the first and higher the second one, so that to obtain a very quick reponse of the system without generating instability problems.

With this solution, that can be adopted also with the friction wheels solution, it is possible to obtain very high performances in operation without wind.

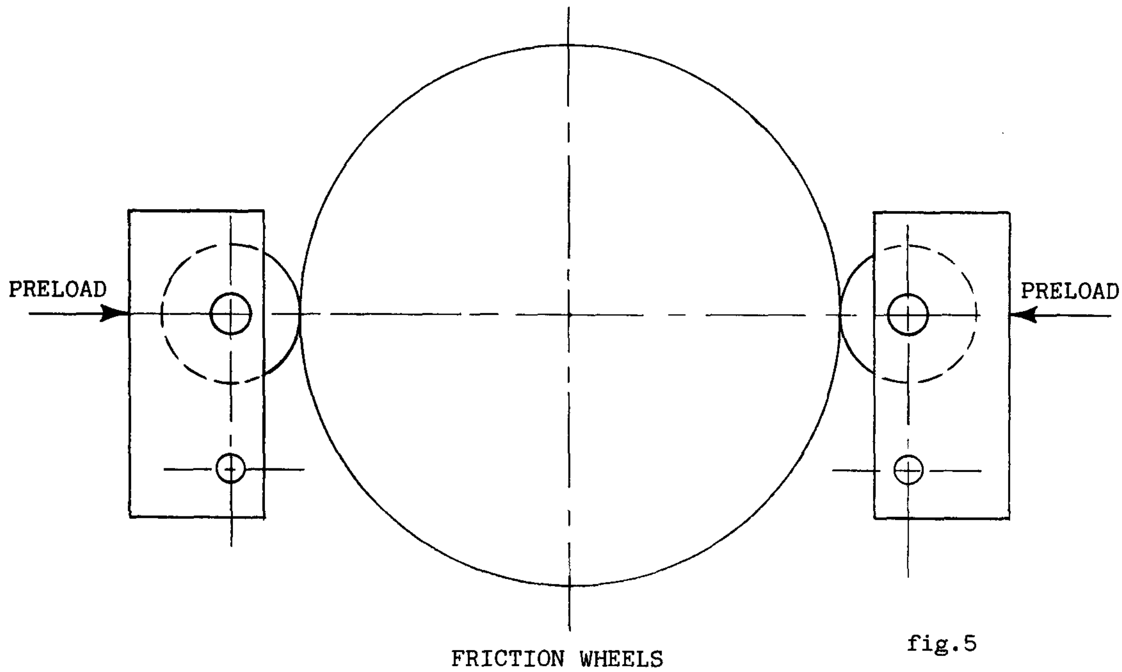

fig.5

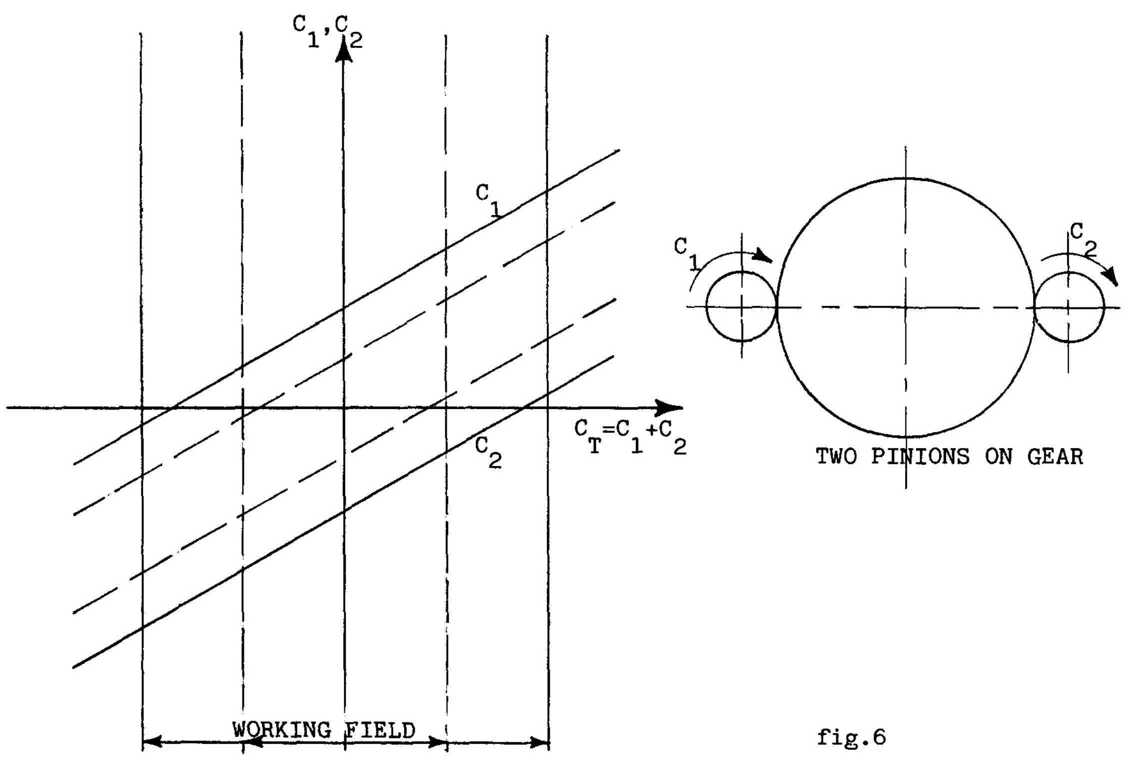

fig.6

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d) Bandwidth

All the three investigated solutions guarantee the requested bandwidth of 3 HZ

Their performance however is different if compared to the wind caused disturbs, that includes important components at high frequency.

The conclusion of the calculations show that with the max torque disturb foreseen, the solution with the torque motor in axis assures the best performances, with errors of 1 arcsec approximately, lower than the ones granted achievable with the other two solutions in the same operating conditions, of 0,2 arcsec.

It must be considered that this little difference in performances is practically negligeable if compared with the sensible structural deformations generated by the wind gusts, that produces temporary pointing errors of about 1 ÷ 2 arcsec

e) Stability

It is possible to obtain it without problems in the solutions with preloaded gears and pinions and with friction wheels, specially because of the relatively small inertia of the load reduced to the motor shaft that allows as a consequence a good separation between the space loop and the speed loop.

In the solution with torque motor in axis, that imposes the introduction of an acceleration loop caused by the presence of an high time constant, the necessary stability can be obtained only with carefull adjustments to be done at the commissioning of the telescope using compensating networks.

f) Tachogenerators noise

This is a very critycal point for the torque motor in axis solution, presenting very elevated figures owing to the low tachogenerator voltage level at the tracking speeds, and can cause transitory position errors of 0,2 ÷ 0,3 arcsec.

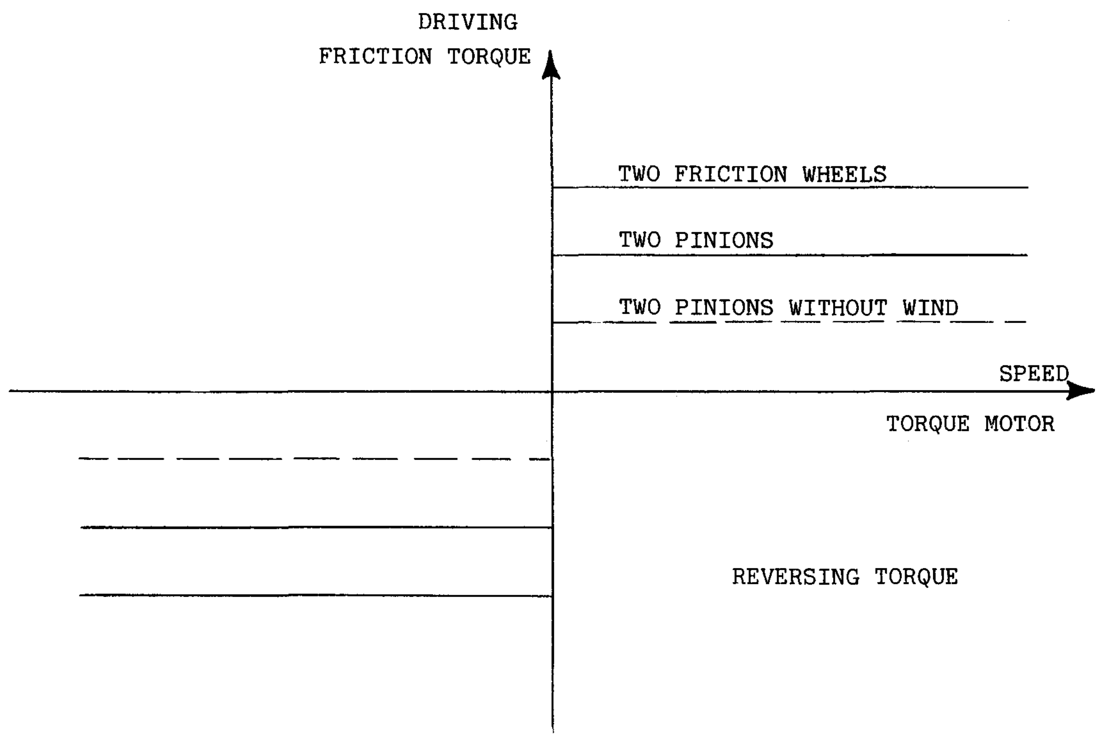

fig.7

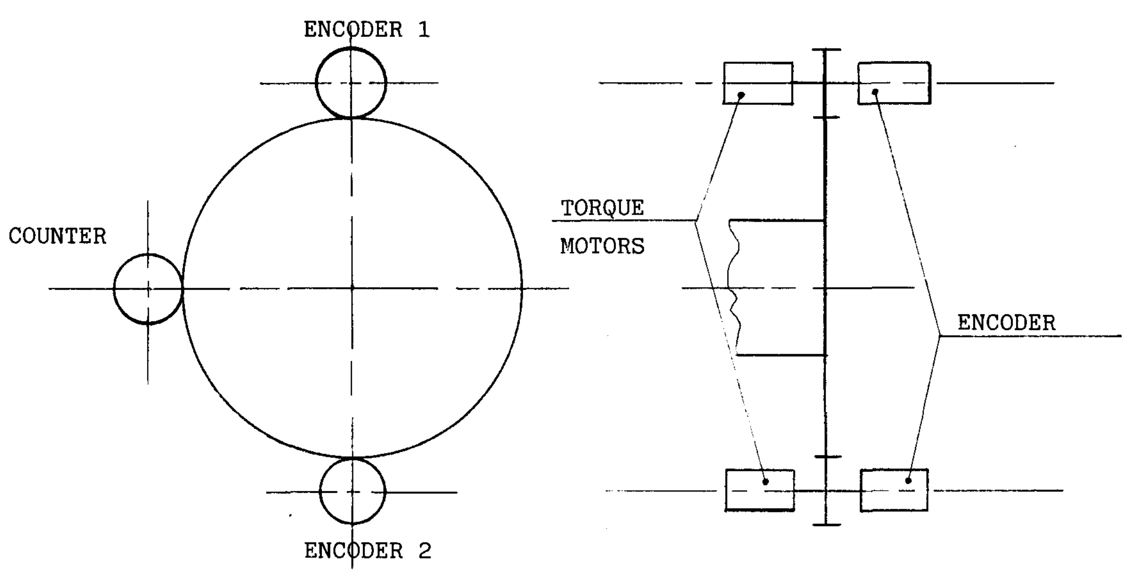

fig.8

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The tachogenerator noise is practically of no influence in the two other investigated solutions.

g) Encoding system

The encoding system must have a resolution so elevated to obtain both a counting frequency more elevated than the bandwidth at the tracking speeds, and together a quantification error of an order of magnitude smaller than the allowed error when tracking.

The requested resolution, taking in account the errors combination on the two axis, is about 0,05 arcsec.
The accuracy can be improved under the own limit of the encoder introducing an adeguate compensation system of the repetable errors, that at the same time compensates the pointing errors caused by the deflections of the supporting structure.

The various alternative solutions investigated are:
- absolute encoder on axis with high resolution and precision
- two encoder with gears in multiplication (see fig.8)
- absolute encoder on axis with high precision and encoder in multiplication with friction wheels

The first system with encoder of hight resolution and precision is very expensive, specially when must be solved dimensional problems of optical beam free prassage on EL axis; moreover there are also limits on the resolution.

The solution with two encoders in multiplication gives an elevated resolution, but the resulting precision dipends from an elaborated system of errors compensation, and from the repeatability of the gears errors.

The system with absolute encoder of high precision, plus an encoder in multiplication with friction wheels, presents a good compromise with an elevated resolution and good precision, that may be improved using a correction system.

CONCLUSIONS

From the comparison of the different performances, the final raccomandation is for the two preloaded gear and pinion in a adaptive system.

The solution with torque motor on axis has togheter the problem of some negative factors of quality, also the problem of using them assembled in couple, as at the present are not on the market single motors with the sufficient starting torque.

Moreover using coupled motors becomes more critical the problem of heat emission in operation, that is more reduced in the systems with two motors and gears and pinion.

The encoding system must be finalized between the two encoder in multiplication, with the absolute encoder in axis, and the absolute transducer in axis and the encoder in multiplication with friction wheels, according to the possibility of finding them.

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