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MODIFICATIONS DONE ON SYNTHESIS GAS COMPRESSOR DRIVE TURBINE

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RESUME

A RCF, Thal, il existe deux unités d'ammoniac de 1350 t/j basées sur la technologie Haldor Topsoe. Les compresseurs de gaz de synthèse de conception Nuovo Pignone sont entraînés par des turbines à vapeur de marque Siemens. Les m/cs ont été réceptionnés en novembre 1984 et février 1985 respectivement pour les deux unités. Différents problèmes ont été rencontrés en cours de fonctionnement. Ils concernaient principalement la stabilité mécanique de la machine, p.ex. la sensibilité au chargement/déchargement, le chargement irrégulier des laveurs sphériques de FBH des turbines HP, la vibration subharmonique de la partie arrière de la HPT et aussi l'instabilité mécanique de 90 % de la charge de l'unité lorsque la turbine fonctionnait à un débit de condensation plus faible. Il en résultait un goulot d'étranglement de production et empêchait une production régulière. Grâce à une série de discussions avec des usines fonctionnant avec des turbines de conception analogue, à l'engineering interne et à des interactions détaillées avec M/s Siemens, on a apporté des modifications à la machine.

La communication traite d'aspects techniques liés aux modifications parmi lesquelles figurent l'installation de plaques durcies en lubroïde sur les supports, un piédestal de soutien dash-II modifié, à la place du piédestal dash-III et l'installation d'un rotor HP superposé raccourci dans les turbines existantes. Les modifications ont été effectuées et la machine fonctionne très fiablement depuis avril 1991. On envisage maintenant de la réviser en avril 1996 après cinq années de marche régulière.

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1. INTRODUCTION

Rashtriya Chemicals & Fertilizers Ltd., a company based in Mumbai (Bombay), India manufactures nitrogenous and phosphatic fertilizers and specialty chemicals at its plants located at Mumbai & Thal. The Thal complex comprises of natural gas based 2 X 1350 MTPD ammonia plants of Haldor Topsoe design and 3 X 1500 MTPD ura plants of Snamprgetti design along with utilities and also other industrial chemicals. The two ammonia plants were commissioned in Nov. 84 and Feb. 85 respectively.

The synthesis gas compressors in ammonia plant are supplied by M/s Nuovo Pignone, Italy for Train-I and by M/s BHEL, India for Train-II. Drive turbine of the machine is designed by M/s Siemens AG; Germany and being the subvendor of M/s Nuovo Pignone supplied the turbine for Train-I. For the second train M/s BHEL manufactured and supplied the drive turbines under technology transfer agreement with M/s Siemens AG. The train is designated as follows:

2BCL 508 - NK 32/36 - EHNK 40/36 - BCL 407 A - 2BCL 407 B.

The suction pressure of the compressor is 27 kg/cm² and makeup discharge pressure is 212 kg/cm². Last barrel of compressor also incorporates a recycle stage and its suction pressure is 209 kg/cm² with discharge at 223 kg/cm² pressure.

The turbine consists of two turbines HP and LP in separate casings and coupled together with a gear coupling. Design speed of the turbine is 10,520 rpm. The HP casing is having inlet steam condition of 104 kg/cm² and 507°C. Extraction from the HP turbine is at 39.6 kg/cm² and 380°C. The steam exhaust is at 0.14 kg/cm² pressure. Extracted steam is utilized for process application and driving various turbines within the plant. Design output power of the machine is 22,420 KW and maximum 30,745 KW at 11800 rpm. Original bearing housing construction of both the LP and HP turbines is of Siemens Dash 3 type. Construction of both the turbines is as shown in the Figure 1.

2. HISTORY

The machine in Train-I was commissioned in Nov. 1984. Commissioning was satisfactory and test runs were also up to the mark. After running the unit for some period lot of problems were found to be existing on the turbine side though all the compressors were performing well. Whenever the units were stopped for problems in other areas of plant, the conditions at the turbines were found disturbed. Sometimes it was not healthy enough to run it. Following problems were faced during these stages of the plant operation:

1. High shaft vibrations
2. High casing velocities
3. Low extraction percentage recovery
4. Inability to raise speed up to the design value
5. Oil leakage from front coupling guard
6. Problems in crossover control valves
7. Lifting of the bearing housing

The matter was reported to M/s Siemens for above problems. After correspondences and interaction from both RCF and Siemens we could arrive at an action plan to overcome these problems and set right the machine to do its job. These action plans are basically divided between short term and long term actions to be taken up and are described below.

3. SHORT TERM MODIFICATIONS

3.1 Oil leakage

The coupling guard design between HP and LP turbines was having overlap on each other with « O » ring as sealing medium. Both the turbines are coupled in such a way that their front sides face each other and so during operation the casings expands towards each other. This inevitably resulted in oil leakage from coupling guard. Being front side these always posed fire hazards. Similarly this design was not flexible enough and limited the bearing housing travel from travelling equally. This contributed to vibrations. A new type of coupling guard was then installed having flexible bellow between both front coupling guards. Due to this oil leakage is completely stopped. Additionally due to flexible bellow axial travel of both bearing housings improved and disturbance of centering in skew fashion reduced. This resulted in lowering vibration level to some extent. Both old and new coupling guards are shown in the Figure 2.

3.2 Improved insulation

M/s SIEMENS suggested improved casing insulation. This involved keeping minimum 4" gap between front bearing housing and casing insulation. This resulted in sufficient air current passing through this gap and heat is restricted from going towards the bearing housing. Additionally a heat shield at rear of the front housing consisting of ceramic fibre blanket layer is provided and both the tierods are covered with asbestos rope. Cooled bearings certainly have better stability.

3.3 Front bearing housing holding down arrangement

Details of the front bearing housing and pedestal of original design is shown in the Figure 4. It can be seen from detail « A » that front bearing housing is resting on four spherical washers which are mounted on the threaded cups fixed on the bearing pedestal. Similarly at detail « B » it can be seen that bearing housing is connected to casing with tie rods. At detail « D » it has been shown that centering guide key is the part of bearing pedestal, where as spherical washers and adjustable bushes are installed in bearing housing.

The tie rod facility is provided only in front bearing housing to take care of axial housing movement with respect to casing expansion /rotor expansion, so as to maintain bearing node point and centering of the rotor. Thus the tie rods push the front bearing housing equally in forward direction during turbine expansion. Bearing housing slides on four spherical washers due to the pushing force of tie rods. This action maintains the vertical elevation of the rotor during expansion. Similarly sideways movement of the bearing housing as well casing is controlled by spherical washers located at guide block of bearing pedestal as shown under detail D.

During running we observed that the front bearing housing is having a tendency of lifting up. All the spherical washers were not getting loaded. Due to this unstable housing condition we were unable to increase the speed of the machine above 10,000 rpm. After 2 or 3 start-ups we were hardly able to maintain 9800 rpm giving plant load limitation. Frequent disturbance in centering due to poor holding down arrangement was resulting in high vibrations. M/s Siemens suggested modification of the front housing holding down arrangement. In earlier design there were two bolts on both left and right side keeping 0.03 mm clearance with bearing housing. Due to convection heat from casing the holding down bolts were expanding and resulting into increase in clearance. This provides room for movement of housing and disturbs the centering. New type of design employs two disk springs in back to back arrangement with 0.80 mm initial compression. Both design are shown in Figures 3 and 4. This provides a constant holding down force on the housing and gives stable condition.

3.4 Modification in crossover LP control valves

The old design LP control valves were not freely operating and at one instance they even got jammed after tripping of the turbine. So also the clearance between valve and spindle always increased. This was resulting into uneven opening of both valves and subsequent poor loading of the machine. Following modifications were done.

1. Jamming of LP control valves was eliminated by replacement of originally supplied Ni-Cu-Fe material sintered bushes with nitrided steel bushes.
2. Gradual damage to the washer between spindle and valve was due to high flow excited vibration. It was eliminated by reducing size of balancing hole on the valve cone bottom.

After the above modifications were incorporated the Mean Time Between Failure of the machine increased. But still we could not run the machine more than 8-10 months at a stretch and rpm of the turbine was also kept limited to 10,000 and at times resulted in bottleneck and break in production. After cooling down the turbine the minimum job consisting of bearings inspection, adjusting the centering and alignment followed. This resulted in at least 10 days of production loss every time. The journal bearing consumption also increased. Every time the bearings were found to be damaged and high bearing inventory had to be maintained. It was also not possible to increase the extraction to more than 55%. To meet the MP steam requirement HP steam had to drawn through P.R.D.S. and had resulted in unproductive expenditure of energy. Increasing the extraction above this value resulted in high radial vibrations. Keeping the machine running on higher vibrations and disturbed centering was damaging the front steam gland. This leaky steam was increasing the vibrations still further. After three such minor overhaul we had to go for major overhaul just after three years of running. It was also disturbing our planned major shutdown for the plant.

4. ANALYSIS OF THE BREAKDOWN/PROBLEM

During the major overhaul of turbine, the following observations were made:

1. Centering was getting disturbed maximum in up down and sideways fashion.
2. Deep grooves were observed on the seating face of the front bearing housing at the spherical washer points.
3. Deep grooves were also observed on sliding faces of the center guide key of front housing.
4. Front steam gland sealing fins were getting damaged unevenly.
5. Some rotor fins were also found damaged.

Material of construction of bearing housing is softer than spherical washer. During running and in frequent start-ups and shutdowns (including those for process reasons also) vibrations resulted in pitting at contact surface. Continuous pitting results into formation of grooves. Once these grooves are formed travel of the bearing housing is not uniform. Due to this rotor center is always disturbed. The threaded supports are also prone for sinking and disturbs the rotor center. Running the machine in this condition resulted into damaging the bearings and the fins of front steam gland. Due to leakage steam hitting on the bearing housing, the problem aggravates. Turbine shaft vibrations and casing velocities remain on higher side and result into rpm limitation and less production.

Problem was studied by M/s Siemens and RCF jointly with a view of increasing the mechanical stability of the machine. After long deliberations and interactions, two courses of action were decided. One is a short term plan to improve the front bearing housing support system. Secondly mid term planning was considered where it was decided to go in for some changes to increase the rotor stability. To overcome the above problems following modifications were required to be done under mid term planning.

5. MID TERM MODIFICATIONS

5.1 Modification of front bearing housing support

1. Fixing of hardened plates on the seating surface of front bearing housing to increase the wear resistance. Replacement of spherical washers by lubrite plates to increase the load carrying area.

Lubrite plates are plates having graphite pockets in it. Graphite being a solid lubricant helps smooth travel of the bearing housing during running. With the modified holding down bolts, the rotor center is maintained.

2. Fixing of hardened « L » shape plates on the center guide key to increase the wear resistance and to help easy guide. The required spares were obtained from M/s Siemens in time and we waited for the opportunity to implement it. Similar machine in train-II is by M/s BHEL. During November 1989 it had high vibration problem and it was decided to overhaul the same. As we were having modification spares in hand we decided to implement it on this machine. The modifications are shown in Figure 5.

Following activities were involved while doing the modification of front bearing support system:

1. Opening of turbine top casing, bearing cover and removing of stationary blade carriers, bearings, glands, couplings, etc. as is usually done in normal overhaul.
2. Support the bottom casing at front on a « I » beam to take the load. Provide two jack bolts on the « I » beam. Jack these bolts so as to lift the casing by 0.05 mm. Release the jackbolts normally provided for the bottom casing. Make sure that there is no deflection of the casing. Now load is transmitted to « I » beam.
3. Remove the bearing housing and pedestal for modification and shift them to workshop.
4. Measure the original thickness of the housing seating face and harden plates as shown. Remove the grooves and machine both left and right faces such that total thickness of it with hardened plates would be approximately same as it was original.
5. Drilling and tapping of the holes on both faces for fixing the plates.
6. Surface contact checking of plates and housing for more than 80% by blue matching and fixing these plates.
7. Check the surface parrallality of plate face to parting plane of housing and correct it if required.
8. Check the trueness of both faces of lubrite plates.
9. Check the surface contact matching of plates with housing and pedestal.
10. Measure the thickness of pedestal center guide key and hardened « L » plates.
11. Remove the grooves formed on the guide key and machine it on both side so that total thickness with « L » plates is approximately same as original. (Similar to the one described above)
12. Machine the seating face of stoppers for « L » plates.
13. Drilling and tapping for « L » plates and stoppers.
14. Surface contact checking of both sides of key with « L » plates and fixing of plates and stoppers.
15. Drilling, reaming and fixing of taper pins.
16. Machining of the area near spherical washers so that lubrite plates can be put and removed without removing the housing. Provide stoppers for these plates so as not to be disturbed during operation by vibrations.
17. Installation of pedestal on the base plate. Check surface parrallality of seating face of lubrite plates with ground and adjust it.
18. Release the jackbolts of « I » beam and take the bottom casing load on the pedestal. Levelling of casing.
19. Put spherical washers and housing along with rotor. The function of these washers was only to adjust rotor center. Complete remaining part of the turbine overhaul as usual. Finally measure the gap between pedestal and housing on both side.
20. Grinding of lubrite plates as per required thickness. Put four dial gauges on parting plane of housing and jack it up equally by 0.10 mm. Insert the lubrite plates on both sides. Release load on the washers.

5.2 Special resources required

1. Horizontal boring machine for machining pedestal and bearing housing.
2. Radial drilling machine
3. Surface grinder
4. Hand angle grinder
5. Surface plate
6. Various drills, taps, reamers, etc.
7. Other tools required for normal overhaul.

5.3 Result of the modifications

After completing the support modification turbine shaft vibrations and casing velocities came down. The rpm could also be increased from 10,000 to 10,300. Earlier it was impossible to increase the extraction above 50%. By doing this modification we achieved 75% extraction at normal level of vibrations. The comparison between old and new health of the machine is as shown in Annexure B.

These modifications made the machine run smoothly for some period. However, after 2 to 3 starts the vibrations and casing velocities started increasing. Similarly extraction had to be limited. After completing two years we had to do bearing inspection and alignment adjustment. We had to do major overhaul after 3 to 3½ years.

Even though turbine mechanical stability was improved after housing support modification machine was unable to reach design rpm and extraction continuously at lower vibrations which was our ultimate aim. To achieve the aim and as discussed earlier for mid term plan a purchase order for new shortened rotor of HP was placed on M/s Siemens.

The old design of HP turbine rotor was having more overhang on both side of the journal bearings. As torque transmission point was quite away from journal bearings, the turbine was unable to take high rpm continuously as after some period bearings started to damage. This often resulted in damage to front steam gland sealing fins too. Increasing the mechanical stability without disturbing the blading and casing was only possible by reducing the overhang portion of the turbine rotor. Hence the shortened rotor concept was suggested by M/s Siemens.

5.4 SHORTENED ROTOR

The difference between old and new shortened rotor is shown in Figure 6 and these differences are as follows:

1. Length shortened at front side is 63.79 mm
2. Length shortened at rear side is 73.68 mm
3. Total reduction in length of the rotor is 137.47 mm
4. Old rotor was having speed pickup probes on the rear side. Due to shortening of the rear side that area was removed; so this gear along with probes was shifted to front side.
5. Due to the shortening of front side of the rotor the mechanical trip cams was removed and had to be provided directly on the front side of coupling hub.
6. Due to the shortening of the rotor on both sides the respective coupling span was increased. New couplings of increased length were procured to take care of this.
7. As a modification, thrust collar thickness is increased from 25 mm to 29 mm to have more strength. New thrust bearing was required for it. New thrust bearing pads thickness is reduced by 2 mm. Also on increasing the collar thickness bearing clearance is increased by 0.05 mm.

Except for the above modifications all dimension of the rotor like journal bearings size and location, span was kept same. The rotating blading and stationary inner casings, etc. were kept identical.

For installation of the shortened rotor of HP turbine following additional spares over and above new design rotor and thrust bearing were obtained from M/s Siemens AG.

1. Threaded plugs for blocking the original holes of speed pickup probes on rear bearing cover.
2. Various spares required for modifying the mechanical trip assembly .

Complete material was made available by Jan. 91 and we took planned shutdown for installing this shortened rotor in April 1991. Overhauling was carried out in presence of expert from M/s Siemens. Activities required for modification of front bearing housing support were same as stated earlier. Apart from it major activities required for shortened rotor installation are as given below.

FRONT BEARING HOUSING

1. Bearing housing cover was sent to workshop for drilling and tapping holes at 600 interval for fixing speed pickup probes. Total three holes were required.
2. New rotor is having mechanical trip cams on the coupling hub. This diameter being more than the original one, the mechanical trip assembly had to be lifted up while fixing. Required plates were available with us. Cam's axial location is same as that of old rotor.
3. Installation of new thrust bearing assembly. Surface contact matching of cage with housing and collar with the pads was done.

4. Front coupling hub of LP turbine rotor was replaced to match HP rotor front hub. New long coupling was installed.
5. Due to increase in the coupling length, oil inlet line for coupling hub was modified.

REAR BEARING HOUSING :

1. As speed pickup probes were shifted to front bearing cover the respective holes of rear bearing cover were blocked by threaded plugs procured earlier.
2. Due to shortening of the rotor on rear side position of the shaft coupling hub got shifted toward bearing side. Coupling sleeve diameter being more than the inside diameter of the bearing housing at that respective place that area was required to be machined. Complete rear bearing housing assembly was sent to workshop and machining was done such that there is sufficient space for free rotation as well as place for lubricating oil supply line to coupling hub.
3. Compressor discharge side shaft coupling hub was replaced to match with turbine rear coupling hub.
4. New lubricating oil line for rear coupling hub was provided to suit for long coupling.
5. New long coupling was installed.

Rest of the turbine overhauling part was same as usual.

The earlier resource and time planning schedule provided for 32 days for the modification of front bearing housing support modifications and installation of shortened rotor. After the completion of the job a comparison of the actual time taken vis-a-vis the plan was carried out. It was seen that in actual practice the job went fairly as per schedule. The break-up of the detailed activity time-schedule sheet is shown in Annexure A at 12 hours per working day at the site and 20 hours at workshop.

The machine was started after completing both short as well as mid term planning and turbine health improved remarkably. Comparison of the turbine health between old design and new design is as given in Annexure C.

Machine is running continuously without any trouble to-date. In April 96, it completed 5 years and still the health is good which can be seen from following sample readings.

Turbine speed : 10415 rpm
Shaft vibrations Front/Rear : 25 / 20 Tm
Front housing velocity H/V/A : 2.4 / 3.5 / 1.2 mm/s
Rear housing velocity H/V/A : 4 / 1.8 / 2.6 mm/s
Plant load : 100 %

By doing the above modifications not only availability and plant load has increased but our spare parts consumption has been reduced by 25%. Though our synthesis gas compressor turbine is performing well we planned our major plant outage for upgrading the capacity and took this opportunity to overhaul this turbines as well as all driven compressors in April 96. We have also done similar modification of front bearing housing support in process air compressor drive turbine of 12,300 KW and results have been excellent.

6. LONG TERM MODIFICATIONS PLANNED

These modification improvements are planned to further improve the bearing system based on the Siemens Dash II design.

6.1 Support of casing

Hitherto the upper part of casing was resting on the bed plate under front bearing housing. This system needs the support of the lower part of the casing on threaded cups. Subsequent to complete assembly of turbine the load is shifted on the threaded cups support through upper part of casing. This resulted in centering disturbances as threaded supports are prone to sinking. During the opening of casing similar problems are faced. To avoid this the support of casing is proposed to be shifted to the bottom part of casing. Bottom casing rests on front bearing pedestal through spacer plate.

6.2 Adjustable brackets for journal bearing

The present design does not have this facility. When provided with this facility with shims there would be ease in centering. Once centered, no disturbance will take place like the ones with earlier design where the load is taken by threading.

6.3 Travel of the housing

In earlier designs, the casing expansion is transmitted to the bearing housing through tie rods. Non-uniform heating of these rods causes uneven expansion in both the rods, ultimately resulting in misalignment of the rotor bearing system. New design is proposed to adopt tongue and groove locking arrangement between casing and the front bearing pedestal.

Designs in the past incorporated spherical washers through threaded cup back-up for guiding the front bearing housing during axial movement. The new design would incorporate a key. The front bearing pedestal of the new design slides over frictionless lubrite plates as shown in Figure 7.

The above long term action plans are as yet in the nascent stages and shall take a better shape in near the future and shall be implemented in next major overhaul of machine by procuring newer bearing housing, pedestals and connected parts.

7. ACKNOWLEDGEMENT

We would like to thank our organization M/s RCF for having given a chance to us, to be associated with critical modifications on a major machine. We also thank the management of our company for allowing us to present this paper to the august gathering of experts, the world over. Finally we would also like to thank M/s Siemens AG and M/s BHEL, India for their total involvement in these modification from concept to implementation and setting right the machine and bringing it up to the mark, ultimately resulting into consistent running and therefore improved production.

ANNEXURE A

ACTIVITY CHART WITH TIME SCHEDULE FOR MID TERM MODIFICATIONS ON TURBINE

Cooling of turbine to 100°C	2 days
Removal of steam exhaust piping	1½ days
Removal of turbine casing after lip seal grinding on steam inlet flanges and casing bolt removal by bolt heating.	3 days
Removal of all stationary covers and rotors	1 day
Removal of both front and rear bearing housings and front pedestal.	1 day
Machining of front housing and fixation of hardenend plates. Machining of rear housing inner diameter for coupling.	3 days
Machining of front pedestal and fixation of « L » plates.	3 days
Installation of pedestal and it's levelling, casing levelling.	1 day
Surface contact check of lubrite plates, pedestal, housing. Installation & levelling of rear housing.	2½ days
Front housing drilling and tapping for speed probe	Parallel
Centering of rotors. Measurement and adjustment of all internal clearances.	3 days
Boxing up of all internals, casings, bolt heating and tightening.	3 days
Installation of all exhauts piping.	1½ days
Final rotor centering, lubrite plate machining and final installation.	2½ days
Alignment of the complete turbo compressor train.	2 days
Boxing up of all bearings, covers and couplings and their guards.	1½ days
Lube oil circulation, insulation of turbine casing and ready for start-up.	1½ days
Total duration	**34 days**

ANNEXURE B

COMPARISON OF TURBINE PARAMETERS BEFORE & AFTER SHORT TERM MODIFICATIONS

PARAMETERS	Before Modifications	After Modifications
Turbine speed RPM.	10,000	10,300
H.P. Shaft vibrations F/R	40 / 60 Tm	25 / 35 Tm
Front casing velocity H/V/A	8 / 8 / 6 mm/s	5 / 5 / 3 mm/s
Rear casing velocity H/V/A	4 / 3 / 3 mm/s	1.5 / 2 / 2 mm/s
Extraction %	55 - 60%	70%
Plant load %	95 - 97%	100 %

ANNEXURE C

COMPARISON OF TURBINE PARAMETERS BEFORE & AFTER MID TERM MODIFICATIONS

PARAMETERS	Old Design Rotor	New Design Rotor
Turbine speed RPM.	10,000	10,450
H.P. Shaft vibrations F/R	40 / 60 Tm	20 / 20 Tm
Front casing velocity H/V/A	8 / 8 / 6 mm/s	1 / 1.6 / 1.8 mm/s
Rear casing velocity H/V/A	4 / 3 / 3 mm/s	1/ 1.2 / 1 mm/s
Extraction %	55 - 60%	75 - 85%
Plant load %	95 - 97%	100 - 105%

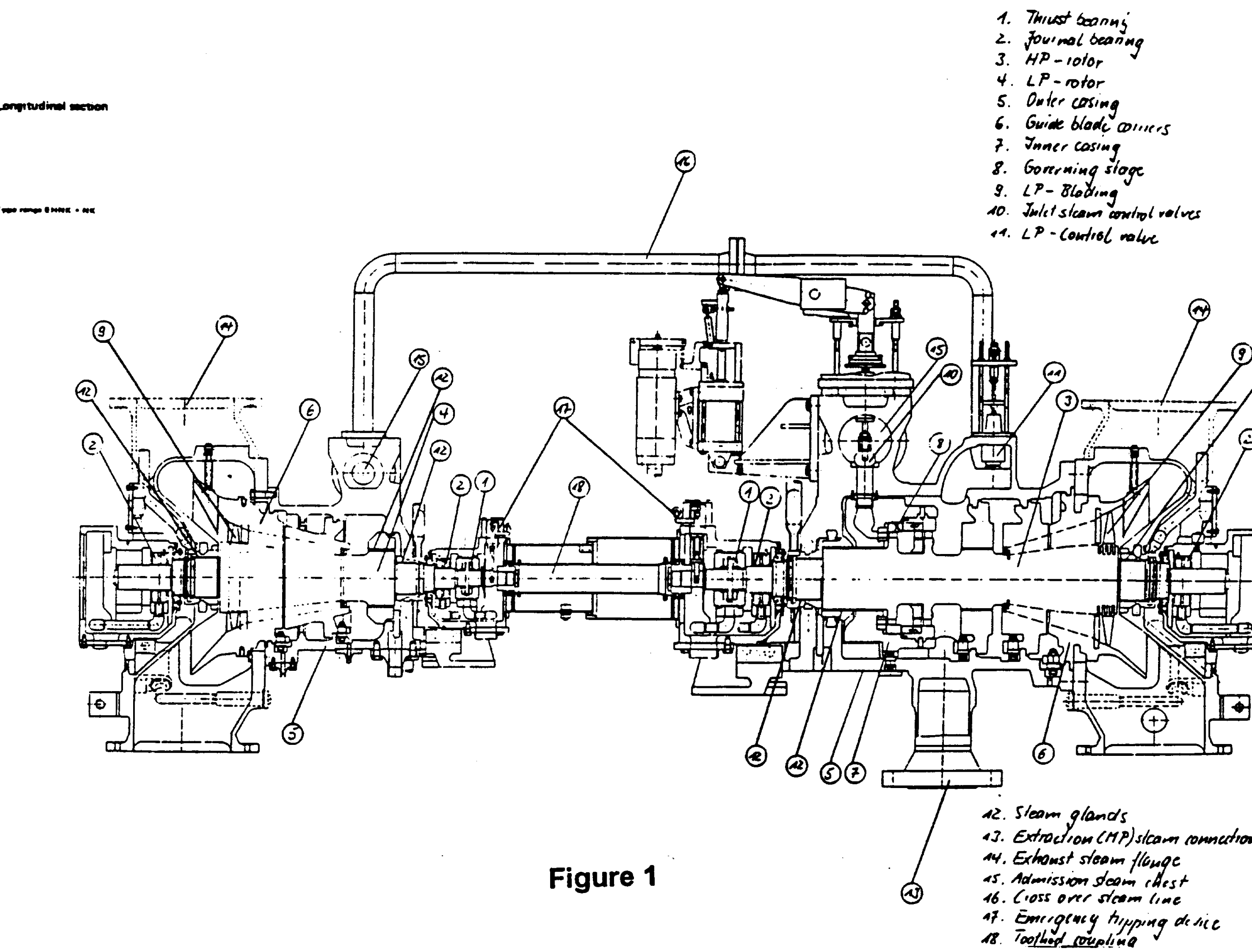

Figure 1

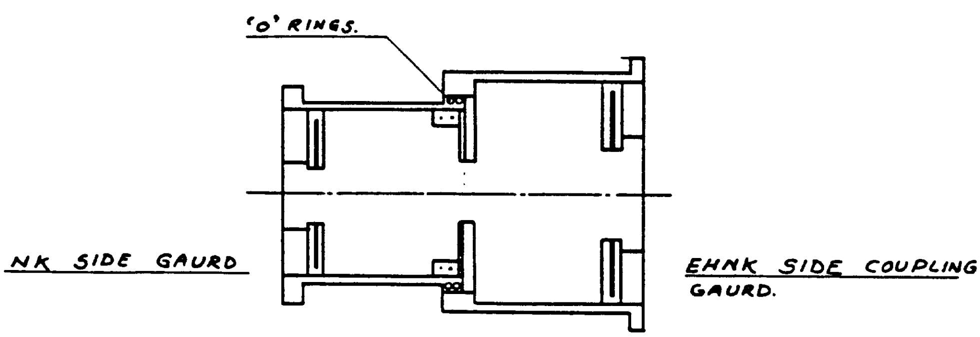

Figure 2 A - OLD COUPLING GAURD

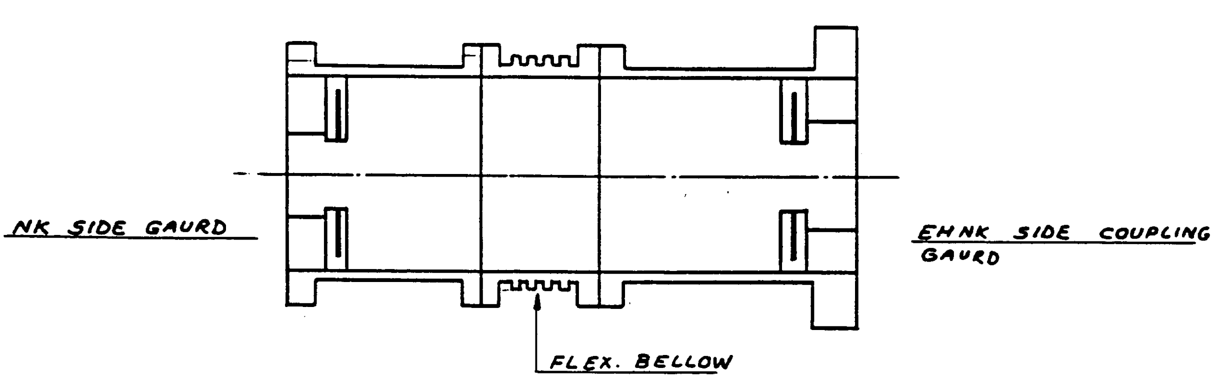

Figure 2 B - NEW COUPLING GAURD

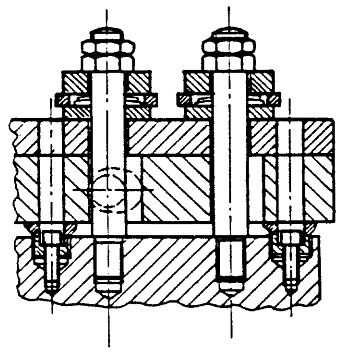

Figure 3 - FRONT HOUSING HOLDING DOWN BOLT MODIFICATION

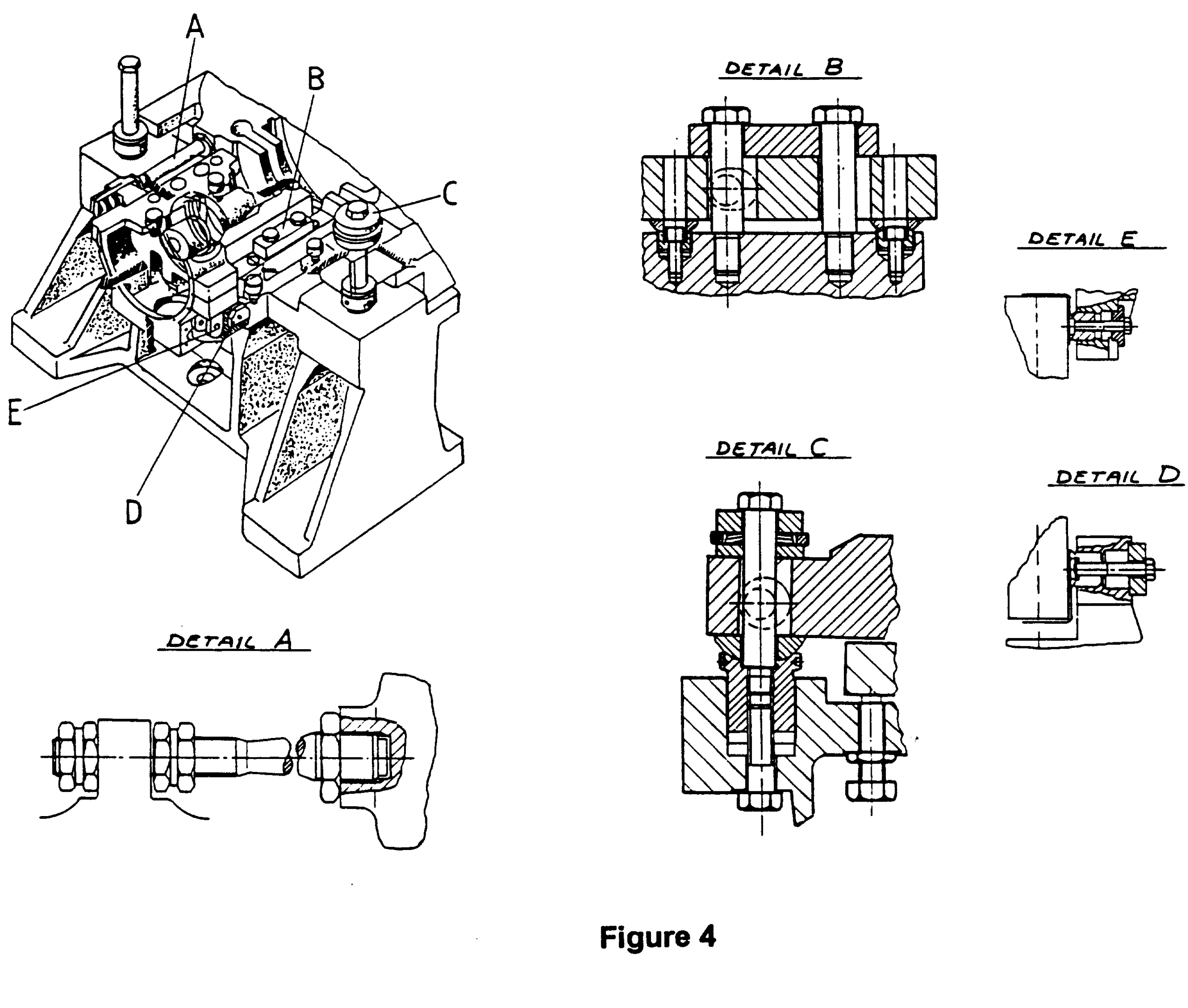

Figure 4

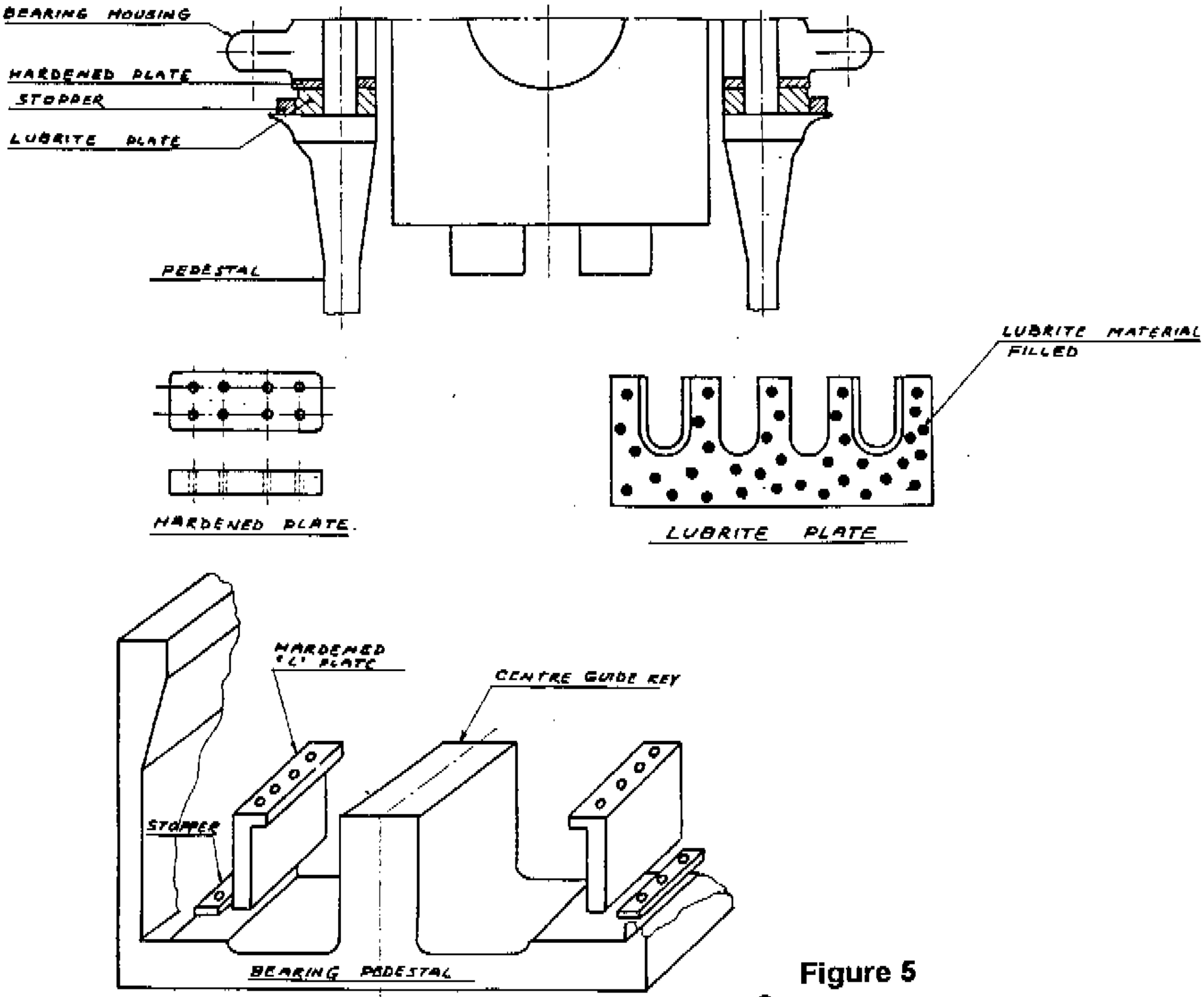

Figure 5

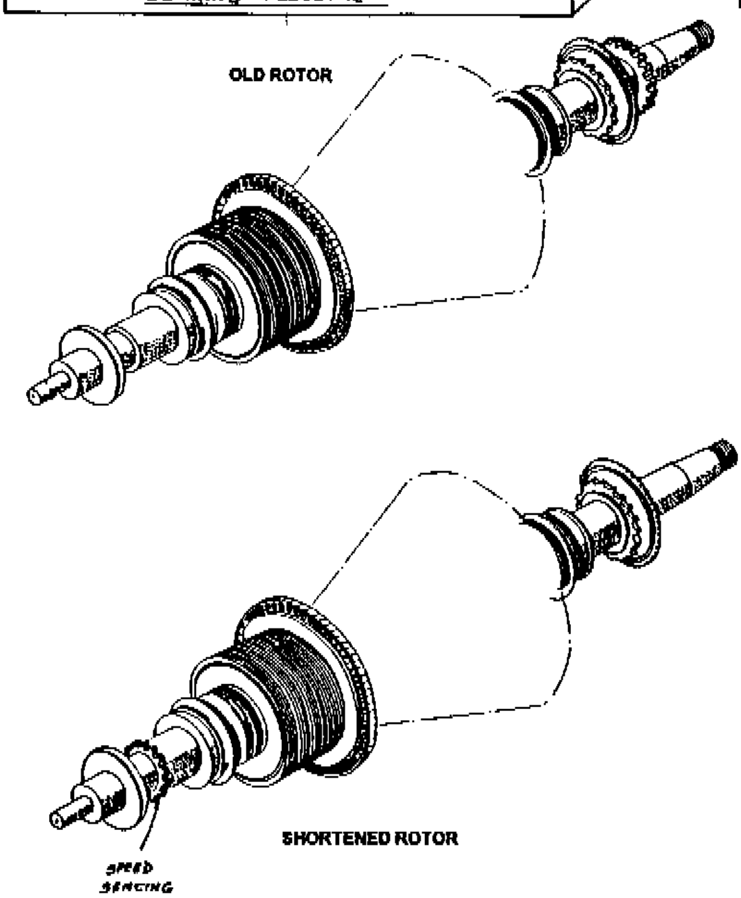

Figure 6

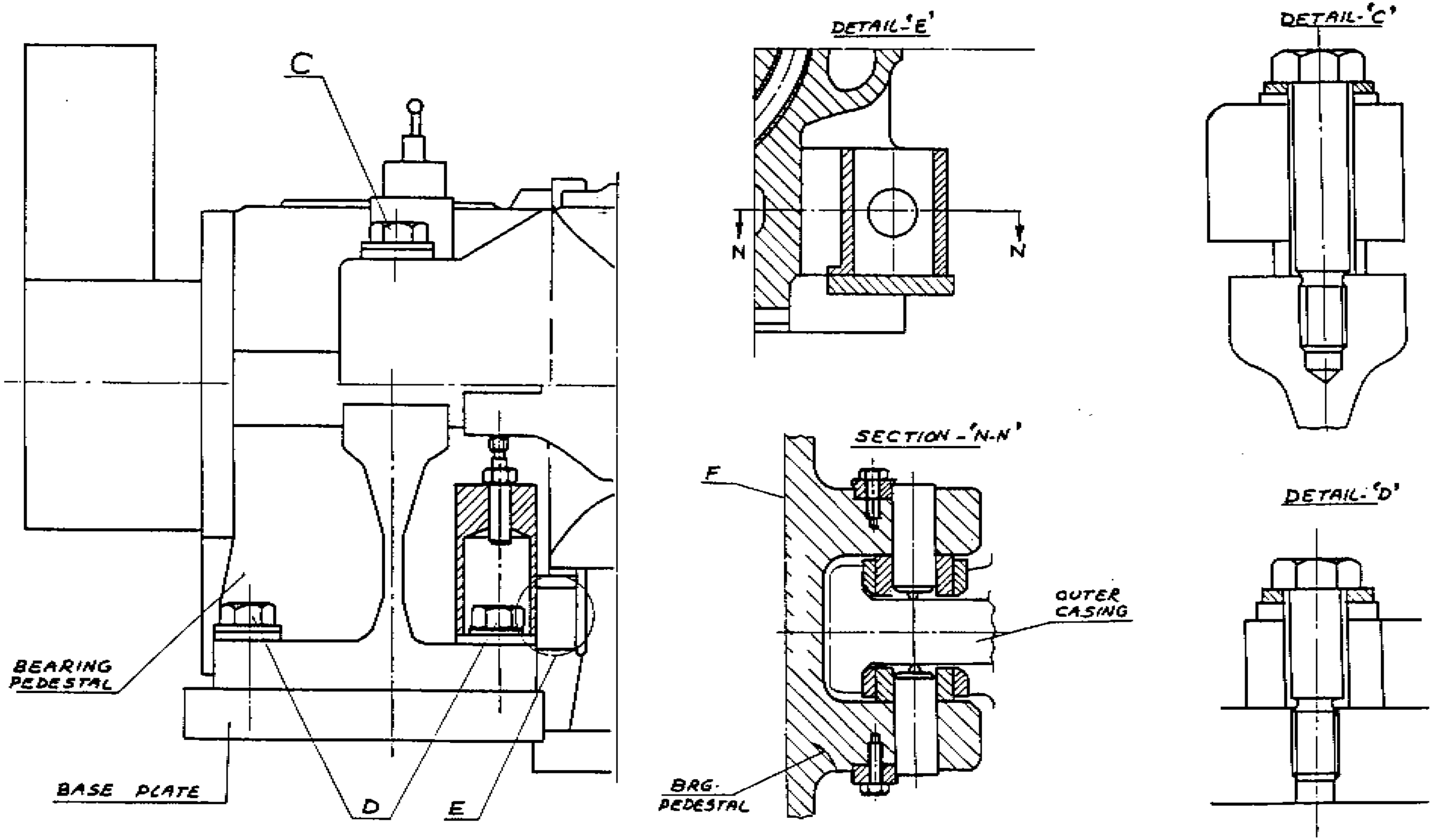

Figure 7 - DETAILS OF FRONT SUPPORTING SYSTEM