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ΠΑΝΕΠΙΣΤΗΜΙΟ ΠΕΛΟΠΟΝΝΗΣΟΥ

ΣΧΟΛΗ ΜΗΧΑΝΙΚΩΝ

ΤΜΗΜΑ ΜΗΧΑΝΟΛΟΓΩΝ ΜΗΧΑΝΙΚΩΝ

ΠΤΥΧΙΑΚΗ ΕΡΓΑΣΙΑ

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ΠΡΟΛΟΓΟΣ

Η παρούσα διπλωματική εργασία εκπονήθηκε στο τμήμα Μηχανολόγων Μηχανικών του Πανεπιστημίου Πελοποννήσου με επιβλέπουσα την κα Δούσμπη Βασιλική. Σκοπός της παρακάτω μετάφρασης είναι η διευκόλυνση της χρήσης του συγκεκριμένου συγγράμματος από όλους του μηχανολόγους, προπτυχιακούς και μεταπτυχιακούς φοιτητές καθώς επίσης και από φοιτητές ERASMUS που φοιτούν ή πρόκειται να φοιτήσουν στο τμήμα αυτό. Επιπλέον, ένας ακόμη λόγος που επέλεξα τη συγκεκριμένη μετάφραση ήταν το γεγονός ότι το βιβλίο δεν υπάρχει μεταφρασμένο στην Αγγλική γλώσσα. Επομένως, θεώρησα σκόπιμο να κάνω τη μετάφρασή του καθώς αποτελεί ένα σημαντικό εργαλείο για κάθε μηχανολόγο μηχανικό που επιθυμεί να ασχοληθεί με τον τομέα της συντήρησης των περιστρεφόμενων μηχανών, αφού παρουσιάζει αναλυτικά τους τρόπους συντήρησης των μηχανών και πρόληψης των βλαβών. Κατά τη διάρκεια της μετάφρασης ακολουθήθηκε πλήρως η διάταξη και τα περιεχόμενα του συγγράμματος. Αρχικά, μελέτησα την αναγκαιότητα της συντήρησης των μηχανών και τη σημασία της για την ομαλή και αποδοτική λειτουργία τους σε μια γραμμή παραγωγής. Στη συνέχεια, είδα αναλυτικά τις μορφές των βλαβών και τους τρόπους συντήρησης, τα μέρη της μηχανής στα οποία είναι πιθανό να εμφανιστεί η βλάβη, αλλά και τους τρόπους επίλυσης των προβλημάτων που προκύπτουν.

Τέλος, θα ήθελα να ευχαριστήσω θερμά την κα Δούσμπη Βασιλική για την πολύτιμη βοήθειά της και την ορθή καθοδήγησή της κατά την υλοποίηση της διπλωματικής.

<u>**Υπεύθυνη Δήλωση Φοιτητών:**</u> Η κάτωθι υπογεγραμμένη Φοιτήτρια έχω επίγνωση των συνεπειών του Νόμου περί λογοκλοπής και δηλώνω υπεύθυνα ότι είμαι συγγραφέας αυτής της Πτυχιακής Εργασίας, αναλαμβάνοντας την ευθύνη επί ολοκλήρου του κειμένου εξ ίσου, έχω δε αναφέρει στην Βιβλιογραφία μου όλες τις πηγές τις οποίες χρησιμοποίησα και έλαβα ιδέες ή δεδομένα. Δηλώνω επίσης ότι, οποιοδήποτε στοιχείο ή κείμενο το οποίο έχω ενσωματώσει στην εργασία μου προερχόμενο από Βιβλία ή άλλες εργασίες ή το διαδίκτυο, γραμμένο ακριβώς ή παραφρασμένο, το έχω πλήρως αναγνωρίσει ως πνευματικό έργο άλλου συγγραφέα και έχω αναφέρει ανελλιπώς το όνομά του και την πηγή προέλευσης.

Η Φοιτήτρια

Γεωργοπούλου Ειρήνη

ΠΕΡΙΛΗΨΗ

Το παραπάνω βιβλίο χωρίζεται σε επτά κεφάλαια, το καθένα από τα οποία παρουσιάζει και αναλύει διαφορετικά στοιχεία για τη διάγνωση βλαβών και τη συντήρηση των μηχανών. Το πρώτο κεφάλαιο εξηγεί την αναγκαιότητα της συντήρησης για τη σωστή λειτουργία των μηχανών και αναλύει τα ερωτήματα για τον προγραμματισμό του ελέγχου τους αλλά και τα είδη συντήρησης και τους τρόπους προγραμματισμού, οι οποίοι αφορούν σημαντικά εκτός των άλλων και τις ενέργειες της διοίκησης. Στη συνέχεια, στο κεφάλαιο 2 γίνεται εισαγωγή στην τριβή, η οποία αποτελεί πρόβλημα για τις μηχανές και εξηγείται η λίπανση και τα διάφορα είδη της αλλά και οι ιδιότητες των λιπαντικών. Το κεφάλαιο αυτό ολοκληρώνεται με τα βιομηχανικά λιπαντικά.

Το κεφάλαιο 3 πραγματεύεται την αναγκαιότητα των εδράνων κύλισης για τη στήριξη των ατράκτων και των αξόνων. Δίνονται αναλυτικά οι τρόποι στήριξης αλλά και ο τρόπος υπολογισμού και εκλογής των ρουλεμάν. Επιπλέον, αναλύονται οι αίτιες βλαβών τους και οι λύσεις αλλά και ο τρόπος διατήρησης της αξιοπιστίας τους. Στο κεφάλαιο 4 γνωρίζουμε τα είδη των οδοντωτών τροχών καθώς και τα απαραίτητα στοιχεία που αφορούν την εκλογή τους, το κόστος και τα συστήματα προστασίας από φθορές. Κατόπιν, γίνεται αναφορά στα υδραυλικά συστήματα, στο κεφάλαιο 6, σχετικά με τον τρόπο λειτουργίας τους, την εγκατάστασή τους και παραμέτρους που λαμβάνονται υπόψη κατά την εκλογή τους.

Τέλος, τα κεφάλαια 9 και 10 παρουσιάζουν τα πλεονεκτήματα και μειονεκτήματα των ειδών συντήρησης και την τεχνογνωσία που είναι απαραίτητη προϋπόθεση για τη διάγνωση και συντήρηση. Επιπλέον, μέσα από διάφορες μεθόδους και συσκευές υπολογίζεται η συχνότητα των βλαβών και γίνεται η διάγνωσή τους. Ακόμη, γίνεται διάγνωση σχετικά με αζυγοσταθμία, κακή ευθυγράμμιση και άλλες βλάβες στα επιμέρους εξαρτήματα των μηχανών και συστημάτων. Αναφέρεται και αναλύεται η συμπτωματολογία βλαβών και οι τρόποι αντίδρασης των υπευθύνων και αντιμετώπισής τους και τα διαγράμματα των βλαβών και των συχνοτήτων τους.

Chapter 1 defines the maintenance of machines and explains its necessity for their smooth and efficient function. Questions to be asked and analyzed in which critical decisions should be taken about the machine monitoring program (which of them will enter, for how long and how will be monitored, who will undertake each machine, which parameters will be representative, etc.). Subsequently, the types of maintenance are mentioned as well as the planning methods with emphasis to the management actions.

Chapter 2 provides basic facts about the eternal problem of friction that makes this world real (have you ever thought about how our world could be without friction and gravity?). Lubrication is explained for its various types and many facts are listed about lubricants' quality that are useful for the selection of the lubricant about the treatment of wear problems. Industrial lubricants complete this chapter with reference to added ingredients that are used and how they react to materials' wear.

After the introduction and the small historical background to rolling bearings, **chapter 3** explains the necessity of bearings for shafts and axles support. For the different types of roller bearings the ways of their support are given as well as examples of machine design with the objective of reducing errors in the machines and being as close as possible to the point of their optimal operation. The material that follows refers to the method of calculation and selection of bearings, tolerances lubrication in assembly and disassembly and in the procedures and tools used for the operation of the machine without problems. For bearing failures the causes, the symptoms and the solutions are reported in detail. This chapter is completed with a reference to the points that should be noticed on bearings in order to keep credibility and the risk involved.

Chapter 4 reports the types of cogwheels as their selection and installation, the damages of serration, but also the cost of noise reduction and their failures, as well as lubrication systems for protection against wear and overheating.

Hydraulic systems and especially the types of pumps are mentioned in **chapter 6,** where information is given on how they operate, their installation and maintenance, as well as on the design of the hydraulic systems and the parameters that must be taken into account during the selection of accessories.

In **chapter 9** the advantages and disadvantages for every type of machine maintenance are disguised and a step by step training and transfer of expertise for diagnosis and maintenance takes place. The expected fault frequencies in each component are calculated from those previously analyzed. Faults are also diagnosed by the shock pulse method and by a VIB vibration measuring device.

Finally, **chapter 10** identifies damages in mechanical components and diagnosis faults related to balances, poor alignment, looseness and other faults of individual machinery and components, such as roller bearings and sliding bearings. The following table shows the symptoms of the lesions where in each the symptoms are reported and explanations are given for the reaction of the person in charge of monitoring the machine. The chapter ends with the fault and frequency diagrams, in which they are answered as used in the international literature and in the current training practice of technicians.

ΠΕΡΙΕΧΟΜΕΝΑ

Πίνακας περιεχομένων

ΕΙΣΑΓΩΓΗ

Η έννοια της συντήρησης μιας μονάδας παραγωγής γενικότερα περιλαμβάνει:

- Την επιθεώρηση, λίπανση και συντήρηση του εξοπλισμού.
- Τη συντήρηση των κτιριακών εγκαταστάσεων.
- Τη διαχείριση των ηλεκτρικών εγκαταστάσεων.
- Αλλαγές στις εγκαταστάσεις (βελτίωση των παλαιών ή κατασκευή νέων).
- Τη διαχείριση των ανταλλακτικών μηχανολογικού εξοπλισμού.
- Διαχείριση αποβλήτων και ανακύκλωση.
- Την προστασία των εγκαταστάσεων, την ασφάλεια και άλλες υπηρεσίες.

Το εν λόγω βιβλίο πραγματεύεται τη συντήρηση των περιστρεφόμενων μηχανών, οι οποίες μεταδίδουν κίνηση και μεταφέρουν ισχύ για την παραγωγή έργου στις εργομηχανές, οι οποίες εκτελούν εργασίες. Η συντήρηση μηχανών βασίζεται σε πολλά επιστημονικά πεδία και απαιτεί συνδυασμένη προσπάθεια για την επίλυση των προβλημάτων των μηχανών, για αυτό και υπάρχουν πολλές ειδικότητες που συνεργάζονται με τον καλύτερο δυνατό τρόπο στο τμήμα της συντήρησης. Επομένως, την ευθύνη οργάνωσης αυτού του τμήματος συνήθως αναλαμβάνει ένας αριθμός έμπειρων προγραμματιστών ανάλογα με τον όγκο των εργασιών και υποχρεώσεων που πρέπει να διευθετηθούν.

Πιο αναλυτικά, η συντήρηση πρέπει να εξασφαλίζει:

- Οικονομική λειτουργία με τη μέγιστη παραγωγικότητα
- Ανεμπόδιστη λειτουργία με μείωση χαμένου χρόνου
- Βέλτιστο αποτέλεσμα ως προς την ποιότητα
- Πληροφορίες για περαιτέρω βελτίωση του εξοπλισμού και της οργάνωσης

Η συντήρηση μηχανών αποτελεί κρίσιμο παράγοντα για τη ζωή μιας επιχείρησης. Με τον ορο αυτό εννοούμε:

- Επισκευές, κατασκευές, βελτιώσεις
- Διαχείριση υλικών, ανταλλακτικών, εργαλείων και τεχνικών μέσων
- Προληπτικός, προγνωστικός και διαγνωστικός έλεγχος
- Τεχνικός και χρονικός σχεδιασμός εργασιών
- Προληπτικές ενέργειες και αντικαταστάσεις
- Προγραμματισμός των διαδικασιών λίπανσης και εκτέλεσής τους.

Κάποια από τα οφέλη της συντήρησης είναι ότι μειώνει την ποιοτική υποβάθμιση του εξοπλισμού, την άσκοπη επανάληψη των εργασιών συντήρησης, το επενδυόμενο κεφάλαιο, τις υπερωρίες και τους τραυματισμούς των εργαζομένων κ.α., ενώ συμβάλλει στην αύξηση και βελτίωση της διάρκειας ζωής των μηχανών, τη συμμόρφωση σε νόμους και κανονισμούς κ.α.

Στη συνέχεια περιγράφονται οι κρίσιμες αποφάσεις που θα πρέπει να ληφθούν όσον αφορά την παρακολούθηση των μηχανών. Αυτό έχει ως στόχο την ανακάλυψη οσο το δυνατόν περισσότερων ζημιών που πρόκειται να εμφανιστούν και την εξασφάλιση της επίτευξης των στόχων που έχουν τεθεί για τον σχεδιασμό και την υλοποίηση του προγράμματος συντήρησης. Για την ικανοποίηση των παραπάνω υπάρχουν κάποια σημαντικά ερωτήματα που πρέπει να απαντηθούν:

1. Ποιες μηχανές θα πρέπει να παρακολουθούνται.
2. Ποιος θα αποφασίζει ποιες μηχανές θα παρακολουθούνται.
3. Πόσο συχνά πρέπει μια μηχανή να επιθεωρείται.
4. Ποιες παράμετροι θα πρέπει να μετρηθούν, εκτός από τη δόνηση.
5. Ποια είναι τα κριτήρια με τα οποία επιλέγονται και καθορίζονται τα επιτρεπτά όρια καλής λειτουργίας της μηχανής.
6. Υπό ποιες συνθήκες λειτουργίας θα πρέπει να γίνονται οι μετρήσεις.

Ένα εξίσου σημαντικό κεφάλαιο στη συντήρηση των μηχανών είναι τα είδη συντήρησης των μηχανών. Υπάρχουν τρία είδη συντήρησης μηχανών, η λειτουργία ως τη βλάβη, η προληπτική συντήρηση, η προγνωστική συντήρηση και η συντήρηση ακριβείας. Πιο αναλυτικά, η πρώτη μέθοδος συντήρησης είναι ιδιαίτερα απλή καθώς οι μηχανές αφήνονται να λειτουργήσουν χωρίς κάποιος να επέμβει ή να γίνει έλεγχος, εως ότου εμφανιστεί βλάβη ή παραγωγή κακής ποιότητας. Αναφέρεται επίσης και ως Διορθωτική συντήρηση, αφού επεμβαίνει όταν πρέπει να διορθωθούν οι βλάβες και είναι συντήρηση εξ αντιδράσεως, δηλαδή το τμήμα συντήρησης αντιδρά στις βλάβες που εμφανίζονται αντί να τις προλαμβάνει.

Η προληπτική συντήρηση ασχολείται με την ένταξη των διαδικασιών συντήρησης σε ένα συγκεκριμένο πρόγραμμα. Εδώ κάθε σημαντικό μηχάνημα της γραμμής παραγωγής σταματά και ελέγχεται ύστερα από συγκεκριμένες ώρες λειτουργίας, κάθε φθαρμένο εξάρτημα αντικαθίσταται και το μηχάνημα μπαίνει ξανά σε λειτουργία. Η προτελευταία κατηγορία, η προγνωστική-προβλεπτική συντήρηση βασίζεται στην χρήση συστημάτων ελέγχου, τα οποία αποσκοπούν στην ουσιαστική διάγνωση της φυσικής κατάστασης του εξοπλισμού όσο αυτός λειτουργεί. Στόχος είναι η πρόγνωση του χρόνου επισκευής ή συντήρησης πριν την εμφάνιση σοβαρών προβλημάτων.

Η συντήρηση ακριβείας η οποία αποτελεί μια νέα φιλοσοφία στον τομέα της συντήρησης, έχει ως στόχο τον σχεδιασμό, δηλαδή τη διόρθωση των ελαττωμάτων του σχεδιασμού τα οποία πιθανόν προέρχονται από ακατάλληλη μέθοδο εγκατάστασης, λάθος επιλογή υλικών κατασκευής κλπ. Τέλος, σημαντική κατηγορία της συντήρησης αποτελεί επίσης και ο προγραμματισμός της συντήρησης. Ο προγραμματισμός της συντήρησης δεν βασίζεται σε κάποια μεθοδολογία αλλά αφορά την συντονισμένη προσπάθεια όλων των εργαζόμενων σε αυτή, με σκοπό να μην κατασπαταλώνται οι χρόνοι εργασίας σε μη παραγωγικές ώρες αναμονής και καθυστερήσεων.

Εν κατακλείδι, για να πεισθεί η διοίκηση μιας εταιρείας για την επιτυχία του προγραμματισμού συντήρησης και τον συντονισμό και τη συνεργασία των εργαζομένων της προτείνονται πέντε τρόποι, κάποιοι από τους οποίους είναι:

- Μείωση των εργασιών που αναγράφονται στις λίστες αναμονής λόγω αύξησης της ποιότητας των ολοκληρωμένων εργασιών ανα βδομάδα.
- Η ραγδαία αύξηση της διαθεσιμότητας των μηχανών και εγκαταστάσεων στις υπηρεσίες της παραγωγής, οι καλύτερα οργανωμένοι αποθηκευτικοί χώροι, η εκπαίδευση του προσωπικού, κ.α.

Οι Η/Υ αποτελούν βοηθητικό εργαλείο για την συντήρηση των μηχανών. Κάποιες από τις κυριότερες εφαρμογές είναι:

- Σχεδιασμός με τη χρήση Η/Υ γρήγορα και με δυνατότητα εύκολης τροποποίησης του τελικού σχεδίου.
- Καταγραφή των δεξιοτήτων των τεχνιτών για τη χρήση τους από τον προγραμματιστή στη συντήρηση.

CHAPTER 1: INTRODUCTION

1.1 GENERALLY

Generally maintenance in a unit includes:

- Inspection, lubrication and maintenance of the equipment.
- Maintenance of the building installations
- Management of electrical installations
- Changes (improvement of old or new) of the installations
- Management of sewage and recycling
- Protection of installations, security and other services

Here we deal with the maintenance of rolling machines which transmit motion and transfer power for work production in the construction machines, which perform work.

Maintenance of machines is based on lots of science fields and requires a concerted effort to resolve issues arising from the use of the machines. There is a big variety of specialties cooperating in the section of maintenance on industry in order to be beneficial and productive. Consequently, the responsibility of the organization of this department should be assumed by one or more experienced maintenance programmers depending on the volume of obligations of the maintenance department and the number of employees in it. Maintenance work starts from the command to test a machine or a component or is based on the order of priority of the work as it is scheduled. The first thing the maintenance manager does is to examine how the maintenance will be done, who will carry it out and what components will be needed for the repair while examining the history of faults and repairs of the component or machine. For example, if the report says that a technician noticed that the valve is noisy, then the programmer should make sure that it needs to be replaced or repaired and find out what materials are needed for the repair. If some of the materials are not available, orders must be ordered. The repair of the valve is included in the schedule of maintenance work and is done after the warehouse manager confirms that all the necessary materials have been received, so more suitable people must be selected based on the skills required by the repair. Finally, it should be noted that the developer should calculate the time required for each task so that supervisors can better control the procedures.

No matter how technologically developed the production machines are, it is impossible for them to operate and perform, at least at the level they are designed to do, without the necessary supervision and maintenance. Industrial enterprise aims to support the production of products to be produced continuously, with the lowest possible cost and the best quality according to the standards followed. Epigrammatically, maintenance must ensure:

- Seamless operation with reduced time lost
- Economic operation with the maximum productivity
- Optimal side effect quality
- Information for further improvement of equipment and organization

1.2 THE NECESSITY OF MACHINE MAINTENANCE

The cost of maintenance can be very high, up to 35% of business operating costs. By the term maintenance we mean:

- Technical and time planning of works
- Management of materials and spare parts
- Management of tools and productive media generally
- Preventive, prognostic and diagnostic tests
- Preventive actions and replacements
- Planning and execution of lubrication programs
- Repairs, improvements and constructions

From the above it becomes clear that maintenance is not only for repairs, as is generally believed by many, but is a critical factor in the life of the business, related to overall performance. Maintaining the equipment in good working order through maintenance (systematic inspections, detections and corrections of impending failures before they are implemented or before they are disrupted by major disasters) proves that it:

<u>Reduces:</u>

- quality degradation of equipment
- unpredictable faults of the machine
- unnecessary machine repairs
- pointless repetition of maintenance activities
- the quantity of spare parts necessity available
- defects in new machines
- wrong maintenance activities
- disposal of (defective) products
- invested capital
- energy consumption
- overtime and injuries

- the insurance premiums

Increases and improvements:

- machines’ life
- the productivity of maintenance stuff
- compliance with laws and regulations
- the reliability and security

1.3 CRITICAL DECISIONS FOR MACHINE MONITORING

The ability to discover as many impending mechanical failures as possible, without stopping or dismantling machines, requires experience and a data collection system that sets limits, maintains measurement data and carefully selects instruments and devices that are used in measurements. In order of the monitoring program to be able to meet the objectives set, it must follow a systematic work based on the initial design of the program. If we leave the design and implementation of the maintenance program to chance, the goals will not be achieved. For the thorough completion of the maintenance plan there are some specific and important questions to which we must give answers before performing any work. These are the following:

Which engines should be monitored?

This question must be processed from the beginning of the process. Machines that meet one of the following three criteria are generally subject to continuous monitoring:

- If the machine is of great importance for the production
- If the replacement of the machine due to fault is too expensive
- If the machine damage could cause injury or loss of human life

The criteria for the continuous machine monitoring are strictly defined, because the cost of using fixed adapters to some measuring positions that will send their signal through wiring in a control room is quite high. Although a good accelerometer and the control material channel do not have a high cost, however, for enough channels to adequately monitor the machines we are interested in, the cost of cables and system maintenance is often tens of thousands.

On high-end machines it is possible to install more sophisticated machine control programs connected directly to the computer. These systems are based on continuous monitoring techniques with one important difference. The signal which would normally be connected to the control panel here is sent to an electronic device which allows many separate signals to travel to and from the computer following the same path. This electronic device scans every data channel very often. The data is sent to an FFT spectrum analyzer and then to a computer. The computer compares the spectrum data with a pre-installed data base for the specific channel and decides if the data (in any frequency) have exceeded or not the maintenance limits that we have set from the beginning. If the limits have been exceeded then a report is written on the specific problem.

Otherwise, the data is rejected (to avoid overloading the computer memory) and the control goes to the next data channel. The obvious advantage that accompanies such a system is that on the one hand it has all the advantages of continuous monitoring (since the machine is monitored all hours of the day), on the other hand it is possible to easily diagnose faults that are in the early stages and monitor them more carefully.

Periodic maintenance programs are sometimes applied to machines that include the criteria mentioned but more often include the following less important criteria:

- If the repair and/or replacement of the machine is relatively expensive
- If the failure of the machine can cause increased production costs or reduced efficiency of industrial facilities but not necessarily their complete destruction
- If the machine has a bad operating history

Machines that meet the above three criteria very rarely justify spending tens of thousands of euros to monitor them. Nevertheless, the low cost per point that a machine has is something no one can overcome.

Who decides which machines should be monitored?

The decision as to which machines to monitor or type of maintenance should not be entirely the decision of the maintenance manager in question, but should be the result of collaboration between production, maintenance and marketing staff. A logical way to answer the above question is the following:

- The calculation of the quantities and the type of products that will be produced from the specific unit in the coming years should be done by the marketing staff.
- The calculation above will allow the staff members to determine the possible use of each machine for the production procedure.
- Knowing the employment rate of the machine, the operation personnel will be able to estimate the utilization rate of the machine based on its necessity of operation without problems, in order to satisfy the calculation needs for the production of the product.

- The maintenance personnel, knowing the relative reliability of the unit's machines and the probable time required for a repair or replacement of a component of the machine, will be able to generate an approximate reliability index which will indicate the order of priority according to which to be included in the monitoring program. Such a priority number can be calculated as follows:

Priority = usefulness × reliability

Where:

Utility is expressed as a function of how necessary the machine is to achieve the production target.

The reliability is equal to the reverse of the probability of failure of the machine when it operates under the required operating conditions multiplied by the time required to repair or replace the machine.

<u>How often the machine should be inspected?</u>

The interval between two consecutive monitoring of a particular machine depends on its operating history, its design and its operating cycle. Usually this interval is determined more intuitively and taking into account mainly the cost and the difficulty of monitoring each of its measured areas. Every interval between monitoring monitors the risk of any unforeseen damage that will damage the machine. Some logical thoughts for determining the intervals are the following:

- Machines with a poor operating history known defects during downtime or other serious operational complications may need to be monitored daily or weekly
- Most machines may need to be monitored every month or every three months
- Machines with proven long service life or minimal service life may require monitoring every year or every six months
- When a machine starts to have problems, the monitoring interval should be reduced sometimes even to hours until repairs can be made

<u>Which parameters other than vibration should be measured?</u>

The most common values measured are pressure, temperature, flow velocity, spindle rotation speeds (usually obtained from vibration measurements) and power consumption (measured in Watt or Ampere by the motor). The temperatures that develop in the bearings are often an excellent indicator of their poor performance. The method for collecting this data varies from a simple thermometer sufficient for the bearing to a series of thermocouples which are placed as close as possible to the point of measurement.

The only parameters that need to be measured are those that are non-intrusive for the production process. For example if someone wants to monitor a pump, the pressure difference before and after the pump could be easily measured by placing pressure gauges near the inlet and outlet of the pump. However, to measure the flow rate we would probably have to place a flow meter inside the piping, which could cause unacceptable pressure drops in the process.

When deciding what parameters to use to monitor a given machine we must keep in mind that collecting a lot of data will lead to a cumbersome data processing system. If the maintenance manager is unsure of which properties are representative of the condition of a particular machine, he must seek the assistance of the machine manufacturer.

What are the selection criteria for determining the permissible limits of good operation of the machine?

The criteria of efficient operation are easily found either by the equipment manufacturer or by initial specifications given to us when purchasing the machine or by the installation inspection staff. Acceptable degradation of efficiency before a dangerous point that the machine needs to be repaired is a function of process requirements as well as the increased wear that the machine will suffer due to events such as cavitation, obstruction and overheating.

These are three common ways to calculate the good operation of a machine with vibration or vibration measurements.

The first is to set an almost arbitrary allowable vibration level for the whole machine at about 3mm/sec. Although this method is quite simple, it has the serious disadvantage that it is very simple and not only ignores the top low energy faults such as those of bearings but also ignores the whole nature of each individual machine.

A second more acceptable method is to take filtered impulse or noise readings at the measuring positions of the machines and assume that these levels are normal. An increase of 6dB at the level of any bandwidth will be a warning status while an increase of 10dB will be an alarm status. This method has the advantage that it can be implemented immediately. Although some of the faults that are in the initial stage will not be identified, if they preceded the start of the monitoring program, this will be offset by the statistical data that most machines are in good condition at startup and that, due to the simplicity in setting the permissible limits, large number of machines will be monitored.

A third method of determining the permissible limits is to select limit levels based on a detailed analysis of a narrow range of machine. This will allow a competent analyzer to calculate each of the components of the machine separately (each of which generates separate frequencies). This is better than measuring an overall vibration level and hoping that it represents all the critical components (bearings, gears, motors, etc.) of the machine. Every frequency peak must be considered separately and its amplitude level used to determine if the engine's operating mechanism is operating within permissible vibration levels. If not, we should consider the possibility of initial mechanical failure and make a detailed diagnosis of the problem. The disadvantage of this method is the very long time it takes to perform a narrow-bandwidth analysis for each machine and then to integrate it into the monitoring program.

Under what operating conditions the measurements should be done:

There are two operating modes that a machine maintainer can handle. The first and simplest is to study an equipment that operates at a constant speed and with relatively constant loads. In this case the conditions that will be used in setting the permissible limits and in collecting the monitoring data are obvious. Of course, we must keep in mind that even the simplest devices can have fluctuations in their operation between the summer and winter period. Still, since induction motors and engines can show small fluctuations in their speed, any computer-integrated monitoring program must take these changes into account.

If the load or speed in the machine changes due to operating or production requirements then the monitoring of the machine becomes more demanding and we have to decide in which situation we will carry out the tests and measurements.

1.4 TYPES OF MACHINE MAINTENANCE

1.4.1 Operation to fault-Breakdown maintenance

This type of machine maintenance began to be applied immediately with the construction of the first machines. The logic of this maintenance method is very simple. The machines are left to operate without any intervention or control until a fault occurs or poor quality products are produced. Only then is intervention necessary to repair a problem. In the literature it is referred to as "Corrective Maintenance" in the sense that it intervenes only to correct faults and is a maintenance by reaction, in the sense that the maintenance department reacts to the faults that occur and prevents them.

Failure operation does not require significant organization or planning. However, it requires work to be performed under stressful conditions that accompany the occurrence of failure. This maintenance method can be effective when applied correctly. For example, in small and low cost equipment, in equipment where faults can be acceptable (from technical characteristics but also from economic point of view) or in equipment where no other method is possible. In low cost cases due to failure it can be a good strategy and repulsion of problems for the future when they occur.

If a machine is left to operate until the fault, in addition to the significant cost reduction resulting from the unexpected stay of the unit out of operation there is the additional cost of destroying the components associated with the faulty one, which will result in secondary faults of the system.

1.4.2. Preventive maintenance

Many definitions of this method have been established internationally. A common point is to integrate maintenance procedures into a time-bound framework. The logic of this method is the scheduled periodic inspection of the equipment. Every major machine is stopped and thoroughly inspected after specific hours of operation. As preventive maintenance is an interventional method of maintenance, any worn parts are replaced and the machine is put into operation.

Therefore, preventive maintenance consists of a series of activities which are programmed with a frequency dictated by the total time interval from the supply of a machine to its operating hours and its production capacity and:

- either extend the life of a machine (for example, renewing the lubricant in a gearbox prologs its life)
- or reveal that a component is significantly damaged and is about to fail (for example, a three-month inspection showed that there is a crack in the pump seal so finding the crack allows repair before catastrophic damage occurs)

According to this method, maintenance is designed to correct or prevent situations that could lead to damage resulting in loss of production with all the unpleasant consequences. Even if this means that some components may need to be replaced before they run out of reliable operating limits. Although in preventive maintenance the production process stops, the production lost on schedule is much smaller than that which would be lost in an unforeseen breakdown.

The logic behind this maintenance practice is that equipment failure numbers follow a path in which time alone is essentially a factor. Maintenance intervals are predetermined either mainly by the experience of the manufacturer of the specific equipment or to a lesser extent by the systematic record keeping in the installation. In this way, in theory, maintenance procedures can be scheduled at idle times and the necessary spare parts can be ordered at an appropriate time.

The logic of the repair before the fault occurs is the substantial differentiation from the operation of the machine to the fault and the possibility of scheduling the repair and supply time of spare parts. When a component fails, it often destroys the components that come with it, which multiplies the cost of repairing the (total) fault. For example, if the pump bearing is not replaced in time, the fins, the housing and other components will then need to be replaced. Sometimes the damage does not get worse and so the cost of the condition and the cost of the damage are about the same. However, the postponement of the action creates uncertainty and an ever-increasing problem in the maintenance department.

Preventive maintenance requires efficient trained and reliable staff and an organized information handling system that supports the maintenance system with regular scheduled inspections and preventive maintenance work.

An important element of preventive maintenance is the performance of inspections. Inspection is the procedure that:

- examines whether the design or specifications of a machine are the prescribed ones and based on the requirements
- estimates all the factors that can create potential problems
- recognizes all the factors and causes that can lead to termination and estimates the time at which this will happen

Inspections must be scheduled to ensure the smooth operation of the machinery, and in any interventions, repairs or replacements required do not conflict with the production schedule. There are some restrictions on the application of preventive maintenance, such as:

- Failures that do not depend on time, occur randomly and after equal intervals
- Time-dependent failures related to the life of the equipment which cannot be predicted because they do not occur after equal intervals. There are various reasons for this, which are mainly due to the mode of operation and external factors such as poor installation of the component, loss of lubricant, etc.
- The process of shutting down equipment and restarting it each time an inspection is performed. In fact, the bigger and heavier the machines that stop, the more difficult and expensive their restart is.

1.4.3. Predictive maintenance

The crucial point for achieving an efficient maintenance is to find those maintenance techniques that will be appropriate for the activity of each company (industrial, transport, construction, etc.) and will mainly ensure:

- Anticipating impending problems and planning to address them before they become catastrophic.
- Reducing the likelihood of failure in infancy and limiting the effects when it occurs.
- Implementation of a program of quality assurance and continuity of operation of new machines in particular and all mechanical equipment in general.
- Monitoring and recovering of all maintenance parameters so that the collected data is utilized and the conclusions are a guide for action to improve productive activity.

With the aim of the above, which ultimately aim at the gradual shift of maintenance work from restoration and repair work to forecasting, presentation and forecasting processes, predictive or forecast maintenance was developed.

The method of forecasting maintenance is based on the use of measurement and control systems that allow the essential diagnosis of the actual physical condition of the equipment while it is in operation (non-interventional method). The goal is to predict the time of repair or maintenance before the occurrence of serious problems or faults.

Predictive maintenance therefore makes use of the positive features of the two previous methods with their optimal combination to achieve the best possible results. It has the element of prevention in the occurrence of the fault but uses the prognosis in order to intervene by correcting the fault in time, as in the corrective maintenance, when it is now unavoidable.

This approach has reduced costs compared to repetitive preventive maintenance because maintenance activities are performed only when justified.

The implementation of a system of forecast maintenance requires good organization and the infrastructure of workshops which, however, are not separated into a team of control and interventions. They are divided and decentralized into smaller areas of responsibility that perform all controls and interventions.

A program is followed which arises in collaboration with the production managers for the best possible exploitation of the operation of equipment. The condition and performance of the equipment is constantly monitored dynamically (condition monitoring).

Most machine checks are done during their operation and even with their maximum load. The resulting data provide information about the condition of the machine and help predict the intervention time for maintenance and correction. Only when the repair is scheduled its operation stops.

The ultimate purpose of forecast maintenance is to carry out maintenance work at a scheduled time before the equipment fails in operation and when maintenance is economically justified, that is when its cost does not exceed what the equipment failure would entail.

While the philosophy of preventive maintenance is more about time-dependent failures, preventive maintenance deals with random and sudden problems, which it tries to identify and correct in a timely manner. Although failures cannot be completely controlled, by adopting this maintenance method the accidental failures and their effects can be significantly reduced.

1.4.4. Prediction maintenance-precision or design out maintenance.

Precision maintenance is a new maintenance philosophy that has been developed in recent years. It is oriented to the design, that is, it aims to correct design defects, which may arise from improper installation method, incorrect choice of construction materials, vague definition of operating specifications, etc. Obviously, this is a technical problem but the department is still responsible for maintenance. For this reason, the interaction of the maintenance and design departments is required to a large extent so that the maintenance engineer works closely with the design engineer.

In some cases it is either possible to find a routine maintenance activity that ensures the desired level of equipment availability or it is impractical to carry it out with the required frequency. However, even in these cases, something must be done to reduce the risk of multiple failures to a tolerable level. In these cases it becomes necessary to reconsider the design. The redesign mainly concerns some interventions in key points of the machine and the replacement and selection of other alternative solutions. If the failure has only economic consequences, then the need for redesign is assessed on the basis of economic criteria.

The logic of this method is different from the rest. While most maintenance methods aim to eliminate failures or the effects of failures, precision maintenance aims to address failures by eliminating the cause of the faults.

Precision maintenance aims at the heart of reliability by improving design imperfections. It has an advantage over other maintenance methods in that it is applied only once to bring the desired results. Its aim is to eliminate the causes that cause a decrease in reliability and not the effects of these causes. There are two requirements for implementing precision maintenance, as follows:

- It is necessary to have a management team that believes in innovation and follows the market. Without market innovation is not possible.
- The implementation of precision maintenance requires competent and experienced researchers who are provided with the time and capital to make the analysis/synthesis that will lead to the desired improvements. Researchers need to know in depth the laws and principles of engineering to solve technical problems.

1.4.5. Maintenance scheduling

Care must be taken not to misinterpret the concept of “Maintenance scheduling” as there is a risk of confusion with preventive maintenance. Although preventive maintenance is based on a work schedule defined by the service life of the components, maintenance scheduling does not introduce any methodology or implementation technique, but only the coordinated effort of all employees in it, so that times not to be wasted on useless non-productive hours of waiting, delays, etc.

The fight against dead times, to win the non-productive staff, is of the utmost importance. Indeed, unless a maintenance manager is appointed by management, we will have many human-hours wasted on waiting times for tools, materials, parts, spare parts, operating instructions, transportation, etc., with an average of their working time in this way. Thus remaining productive time of 30% of its total working time. After the implementation of the maintenance program, the productive time increases significantly and this strongly affects the labor costs of maintenance. If for example we accept that in order to produce a job in which the program has been defined in advance and 10 craftsmen are required who work 60% of their time productively, while without a program they would work 30% productively, then without planning would be required:

$$10 \text{ people} \times \frac{60\%\ productive\ to\ total\ time}{30\%\ productive\ to\ total\ time} = 20 \text{ people}$$

That is, we see that by employing a responsible maintenance planning we save 10 people. The total amounts that are unnecessarily spent annually on wages and overtime are large.

In the case of part-time jobs, developers know that what craftsmen need to get started is basic concise instructions, while supervisors need time for individual tasks. In this way, emergency work can begin without significant delays. After all, it is obvious that compared to the other advantages of timely scheduling of maintenance work, the calculation of individual and total times is its most important contribution.

With the implementation of the scheduled maintenance large improvements in the speed of execution of the works have been observed. For example, a power plant managed to complete two months of pending work in two months. Another feature of the program is that as maintenance becomes more productive the quality of work typically improves, as technicians can make better use of their time which is substantially longer than before and deal with equipment problems appropriately.

Great attention must be paid to the observance of records after each work, to wit the recording of all actions performed on a machine so that in the future they can draw useful information. If for example a new pump cost A euro in trade and the annual maintenance cost of the old one that the factory has is 50% of A, this must be known by the maintenance planners. In this case it would certainly be preferable to replace the old pump. In general, we see that the information must always be recorded in the machine maintenance book, because it is invaluable to the programmer. Unfortunately, most of the time the tasks that need to be scheduled start from scratch, since there are no relevant entries in the files. The book or maintenance file must contain a detailed list, in which each component has a code and whenever we recall it we can read or add data.

The basic maintenance planning steps that must be followed are:

- Checking new equipment maintenance requests. Qualifying for the most urgent of these recent tasks.
- Check the file if the same or similar problem has been encountered in the past and how.
- Determining the appropriate maintenance actions and identifying the components or special tools to be used
- Brief instructions to each craftsman depending on his skills
- Sending the above information to the archive by the program designer to the maintenance staff supervisors to include the work in the department's work schedule

- Send a document to the warehouse managers to place orders or to ensure that the specific components and materials will be available when maintenance begins.

After finishing this, the person in charge of programming, turns his attention to the rest of the maintenance work that needs to be planned and is less urgent. In this type of work one can look for information about the defective part or machine, from the support material provided by the manufacturer, to seek information from relevant literature, etc. (all this mainly if there are no reports in the file of the parts to be maintained).

The reason is that less urgent work offers more time to deal with them. All this information that they discover is correct to record them in the relevant file of the respective component. The maintenance program of an equipment is included in the work schedule of the department by the programmer himself or by a person in charge, who belongs to the same team of planning and maintenance organization and holds this special responsibility.

The way of introducing programming in maintenance is therefore the following:

- Selection of programmers depending on the number of employees (one programmer is recommended every 20 technicians) and their skills (ability to quickly assess the problem and search for any relevant information that is useful from the archives or elsewhere)

- Avoid assigning tasks of supervising the technical staff to the developers. Developers work for the economy of business, we must not downplay their role.

- Frequent and effective communication of maintenance planners with management. It is also the responsibility of the management to inform all staff about the importance of file retention and data collection as well as the basic principles of repetitive maintenance. Managers should not dismiss and discourage staff working in the areas of coordination, communication and training of a company’s departments, because there are immediate negative symptoms to efficiency.

Five ways to convince a company’s management for the success of maintenance planning

- Decrease in the number of jobs on the waiting lists due to the increase in the number of completed jobs per week

- Since well-planned work is more efficient than unplanned work, management wants to increase working hours to scheduled work

- Reduction in first priority job waiting lists (and a high degree of risk to employee safety and production)
- Assurance to management that no increase in maintenance staff (or new hire) will be required in order to address a possible increase in requirements, such as the expansion of the plant's facilities and infrastructure
- Last but not least, the rapid increase in the availability of machinery and facilities in production services, better organized storage, staff training, the presence of supervisors and many other services of the organization and the maintenance planning.

An auxiliary tool in machine maintenance is the PC, because it provides a huge ease of data retrieval that would otherwise be difficult to obtain. The main applications of PC in maintenance are:

- Automation of office work (the most important convenience is the ability to record and manage text files, followed by industrial graphics for managing graphs and maps, e-mail, etc.)
- Design using PC (CAD), fast and with the possibility of easy modification of the final design.
- Ability to manage and calculate the financial elements of maintenance (or even a company, an organization, etc.), as well as the ability to upgrade the old equipment and use additional software after installation on the computer
- Control of the progress of work (e.g. recording of all data-components, maintenance, etc. for all the machines of the unit)
- Recording the skills of the craftsman for their use by the programmer in maintenance
- Task planning (easier when a corresponding task has been done in the past, easy by the programmer of all the data needed from the files, ex-ante valuation of the cost of the work, etc.)
- Staff training (easy access to the computer of all trainers at any time, ease of renewal and modification of the followed maintenance tactics, etc.)

CHAPTER 2: FRICTION AND LUBRICATION

2.1 INTRODUCTION

The lubricant is the working medium during lubrication process that aims to reduce friction and wear during the relative movement of contact surfaces of two cooperating components. Without lubrication no engine movement is allowed. Therefore, lubrication is the insertion of lubricant between the moving parts of a machine with the aim of easy movement and reduction of friction and wear. The lubricants are interposed between the cooperating surfaces and convert dry friction, which has a high loss rate, into wet friction, which has a very small loss rate.

Although the total amount of lubricants used does not exceed 1% of the crude oil consumed from which most lubricants derive, however the effect of lubricants on industry is significant and disproportionately large. Lubricants, in addition to the success of the purpose of lubrication, perform other functions of a protective nature for machinery, such as antioxidant protection, removal of developing heat, working fluids, etc. Lubricants can be solid, semi-solid (semi-liquid), liquids and gases. In industry however, liquids (oils) and semi-solids in the form of fats have prevailed.

Proper lubrication ensures:

Energy savings by reducing friction losses because without lubricants the energy loss would be enormous.

Longer lifespan of components for long-term and smooth operation of machinery.

Reduced maintenance, repairs and spare parts costs.

The force that will overcome the friction is calculated from the relation:

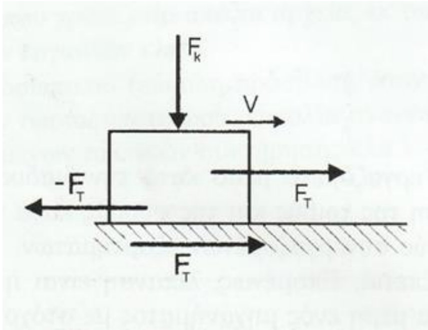
$F_r = \mu \cdot F_\kappa$

Where:

F_r = friction force

μ = coefficient of friction of surfaces. This is exactly where the lubricant intervenes and reduces this factor with the lubrication process.

F_k = force perpendicular to the cooperating surfaces having different velocities.



There are different ways of the lubrication process:

Marginal lubrication.

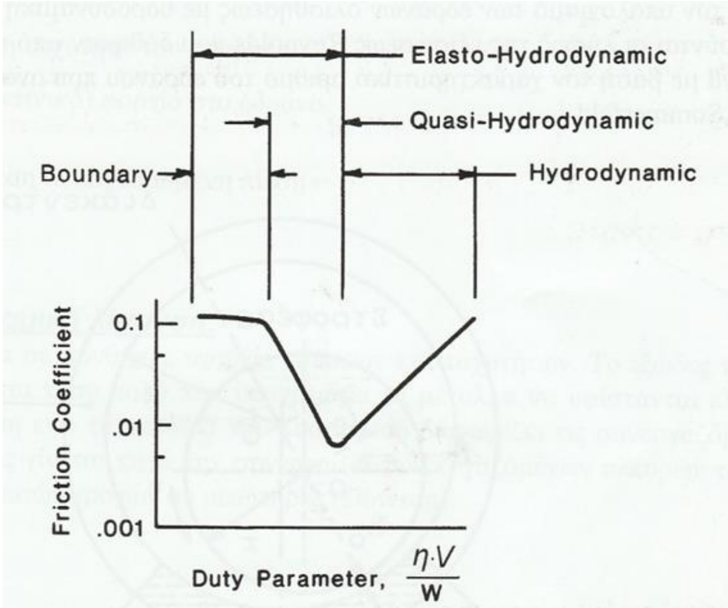
Is characterized by frequent contact of the friction surfaces due to the small thickness (only of a few molecules) of the lubricant. The friction depends mainly on the load and the oiliness of the lubricant. The reduction of wear is achieved with solids of polar molecules of fatty acids that form metal soaps, which by adhesion adhere to the metal surfaces of the components. These substances are added to the original lubricant or formed by the operating conditions and create oiliness of the lubricant. The coefficient of friction in the case of marginal lubrication is between 0.05 and 0.15.

Intermediate lubrication

According to it, the separation of the working surfaces is not complete because the layer of lubricant is not enough. We have periodic contact of the cooperating components and the friction control depends mainly on the properties of the separating surface of the components.

Hydrodynamic lubrication

In hydrodynamic lubrication the lubricant forms a quite large layer and completely separates the moving surfaces. So, there is no contact between the cooperating components. This lubrication shows a small coefficient of friction and theoretically the components have non-contact wear due to their non-contact.



Sketch 2.1 Friction coefficient as a function of Duty Parameter

Force friction is calculated by the relation of Newton-Petroff:

Fτ = η·Uσχετ·A/h

Where:

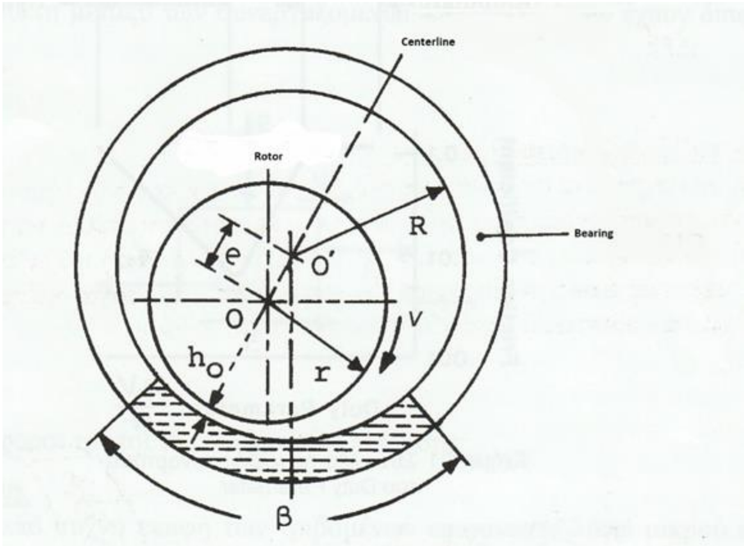
η = the absolute viscosity of the lubricant (N·sec/m^2) whose shear stress in the boundary layer varies with the velocities of the contact components.

Uσχετ = the relevant velocity of the two surfaces (m/sec)

A = the area of the friction surfaces (m^2)

h = the thickness of the lubricating membrane (m)

The student foundation of the theory of hydrodynamic lubrication is due to Reynolds, who in 1886 presented the equations on which the subsequent research on hydrodynamic lubrication was based for the calculation of the sliding bearings with hydrodynamic lubrication. The solutions of the equation Reynolds given by Raimondi and Boyd are used based on the characteristic number of the bearing called S number of Somerfield.



Sketch 2.2 Transverse sliding bearing

$$S = \left(\frac{r}{c}\right)^2 \cdot \frac{\eta \cdot n}{P_m}$$

Where:

β = part bearing arch

η = absolute viscosity of the lubricant

c = radial grace of the rotor-bearing

e = rotor-bearing cam

l = axial length of the bearing

n = rotor speed (shaft)

r = radius of the rotor

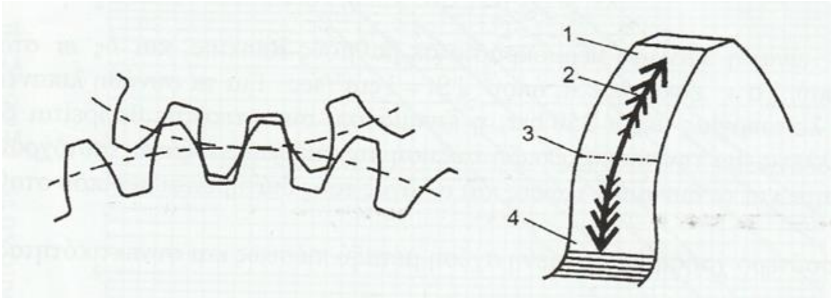
V = rotational speed of the rotor

w = transverse (radial) load on the bearing

pm = w/2rl = average developing pressure

Elastodynamic lubrication

It is created under conditions of high pressures and speeds. The viscosity of the lubricant increases so much that it forces the metals to undergo an elastic deformation while the layer of lubricant separates the cooperating surfaces, as is done during the cooperation of the working sides of the cogwheels in gearboxes (Dowson).



2.2 LUBRICANTS

Liquid lubricants cannot have a shear stress without the presence of motion, regardless of the magnitude of the applied force that creates this shear stress.

The developing shear stress is calculated from Newton's known law:

$$\tau = \mu\frac{du}{dy}$$

Where m is the dynamic viscosity or absolute cohesiveness of the lubricant. This represents the measure of internal friction, which is the resistance of the fluid to flow and du/dy is the change in flow velocity in the direction perpendicular to the flow. The viscosity is usually measured at two temperatures (40 and 100 °C).

It is obvious that low viscosity lubricants move easily through small openings and therefore tend to cause high lubricant losses (leaks) from the sealing points of the systems, while they move easily and with small losses in piping.

High viscosity lubricants, due to increased shear stress, do not easily escape from small openings, but are more difficult to handle.

2.2.1 Consistency of lubricants

Coherence is measured in cP where 1P = 1 Poise = 1 gr cm-1 sec-1 or in other units depending on the unit system used. The most useful in the study of fluid flow is the kinematic coherence which also includes the density ρ of the fluid.

Kinematic coherence changes with the pressure and temperature. For kinematic coherence the effect of temperature is given, according to Walther, from the relation:

$$\log_{10} [\log_{10} (\nu + 0.6)] = b + m \log_{10}T$$

Where T is the absolute temperature in Rankine degrees, b, m fluid constants. V is in cSt where 1 St = 1 cm2/sec. For standard lubricants with operating pressures up to 250 bar, kinematic coherence is considered not to change with pressure, because the increase in pressure simultaneously increases the density and the dynamic viscosity and their ratio is kept approximately constant.

The most used relation between pressure and coherence is:

$$\mu = \mu_t \, e^{ap}$$

Where μ_t is the coherence of the liquid at atmospheric pressure and temperature t, to which the relation applies:

$$\mu_t = \mu_{15°C} \, e^{\lambda(t-15°C)}$$

And for exhibitor **a**, according to Wooster, to the relation:

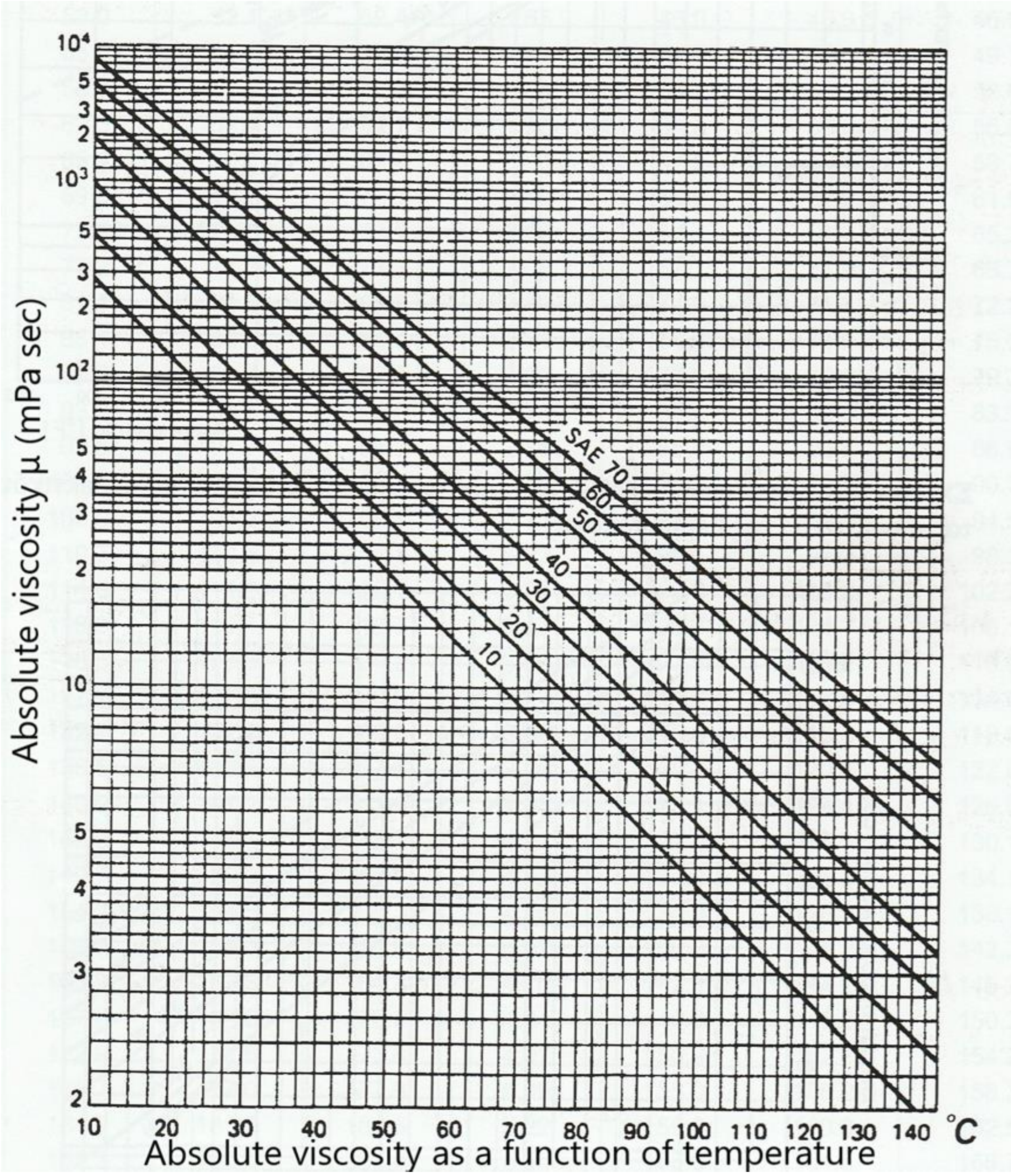
$$\mathbf{a} = (0.6 + 0.965 \log_{10} \mu_t) \, 10^{-3}$$

In which, if μ_t is put in cP, then arises **a** in cm^2/kp. The exhibitor λ depends on the lubricant.

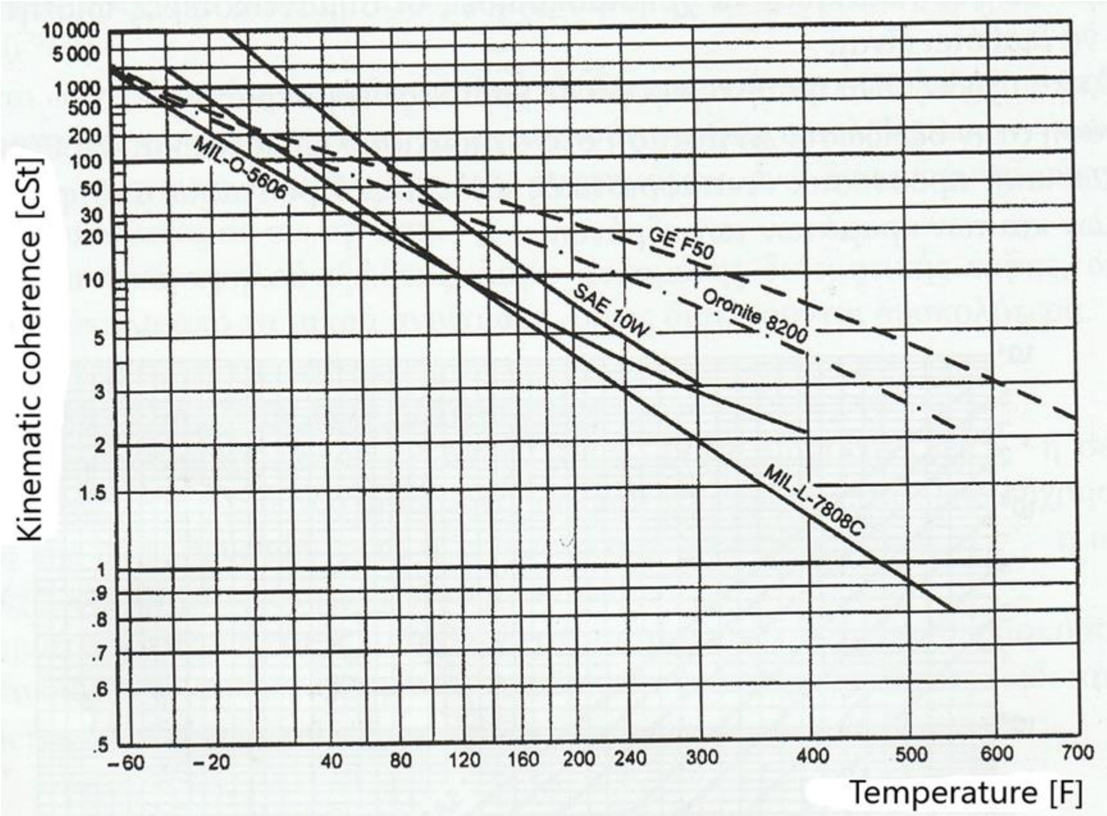
In order for a lubricant to be used, the most important properties it must have are:

Suitable viscosity. Good behavior at low temperatures.

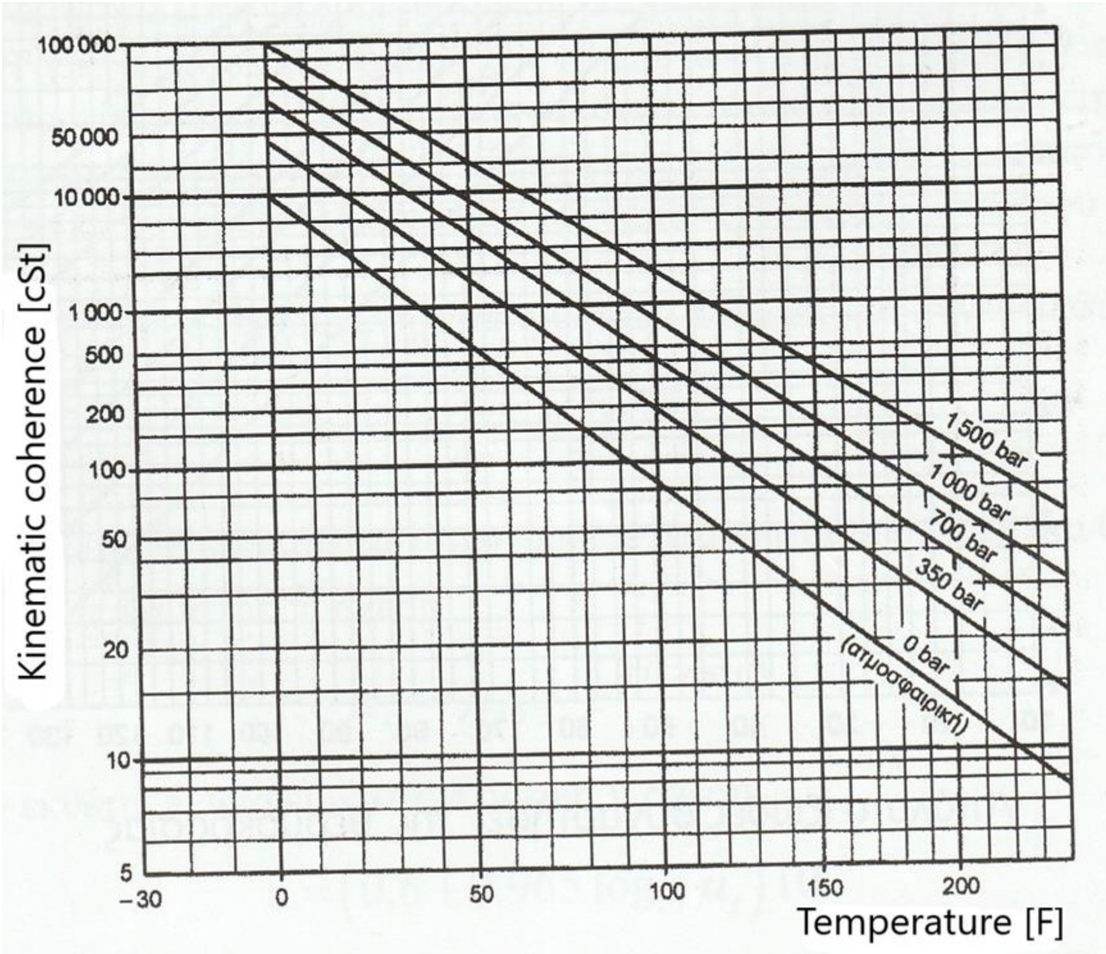
Resistance to oxidation. Resistance to the formation of sediment and deposits. Anti-rust protection. Anti-foaming properties. Protection against wear of metals and bearing alloys.



Sketch 2.3 Absolute coherence change with temperature



Sketch 2.4 Supplementary diagram of kinematic coherence change as a function of temperature.



Sketch 2.5 Diagram of kinematic coherence change as a function of temperature and pressure.

cSt	S.U.S. (Saybolt Universal Seconds)			Βαθμοί Engler	Δευτερόλεπτα Redwood No.1		
	100 °F (37.8 °C)	130 °F (54.4 °C)	210 °F (98.9 °C)	(°E)	70 °F (21.1 °C)	140 °F (60 °C)	200 °F (93.3 °C)
4	39.1	39.2	39.4	1.31	35.3	35.9	36.3
5	42.3	42.4	42.6	1.39	37.9	38.5	38.9
6	45.5	45.6	45.8	1.48	40.5	41.1	41.5
7	48.7	48.8	49.0	1.57	43.2	43.7	44.1
8	52.0	52.1	52.4	1.65	46.0	46.3	46.9
9	55.4	55.5	55.8	1.74	48.8	49.1	49.7
10	58.8	58.9	59.2	1.83	51.7	52.0	52.6
11	62.3	62.4	62.7	1.93	54.8	55.0	55.6
12	65.9	66.0	66.4	2.03	57.9	58.1	58.7
13	69.6	69.7	70.1	2.12	61.0	61.3	61.9
14	73.4	73.5	73.9	2.22	64.4	64.6	65.2
15	77.2	77.3	77.7	2.33	67.7	67.9	68.7
16	81.1	81.3	81.7	2.43	71.0	71.4	72.2
17	85.1	85.3	85.7	2.54	74.4	74.9	75.7
18	89.2	89.4	89.8	2.65	77.9	78.5	79.3
19	93.3	93.5	94.0	2.76	81.4	82.1	83.1
20	97.5	97.7	98.2	2.88	85.0	85.8	86.9
21	101.7	101.9	102.4	2.99	88.7	89.5	90.7
22	106.0	106.2	106.7	3.11	92.4	93.3	94.5
23	110.3	110.5	111.1	3.23	96.1	97.1	98.3
24	114.6	114.8	115.4	3.35	99.9	100.9	102.2
25	118.9	119.1	119.7	3.47	103.7	104.7	106.1
26	123.3	123.5	124.2	3.59	107.5	108.6	110.0
27	127.7	127.9	128.6	3.71	111.4	112.5	114.0
28	132.1	132.4	133.0	3.83	115.3	116.5	118.0
29	136.5	136.8	137.5	3.96	119.2	120.4	122.0
30	140.9	141.2	141.9	4.08	123.1	124.4	126.0
31	145.3	145.6	146.3	4.21	127.0	128.3	130.1
32	149.7	150.0	150.7	4.33	131.0	132.3	134.1
33	154.2	154.5	155.3	4.46	134.9	136.3	138.1
34	158.7	159.0	159.8	4.57	138.9	140.2	142.2
35	163.2	163.5	164.3	4.70	142.9	144.2	146.2
36	167.7	168.0	168.9	4.84	146.9	148.2	150.3
37	172.2	172.5	173.4	4.97	150.9	152.2	154.2
38	176.7	177.0	177.9	5.09	155.0	156.2	158.3
39	181.2	181.5	182.5	5.22	159.0	160.3	162.5
40	185.7	186.1	187.6	5.35	163.0	164.3	166.7

Table 2.1 (continued)

cSt	S.U.S. (Saybolt Universal Seconds)			Βαθμοί Engler	Δευτερόλεπτα Redwood No.1		
	100 °F (37.8 °C)	130 °F (54.4 °C)	210 °F (98.9 °C)	(°E)	70 °F (21.1 °C)	140 °F (60 °C)	200 °F (93.3 °C)
41	190.2	190.6	191.5	5.48	167.0	168.3	170.8
42	194.7	195.1	196.1	5.61	171.0	172.3	175.0
43	199.2	199.6	200.6	5.74	175.1	176.4	179.2
44	203.8	204.2	205.2	5.87	179.1	180.4	183.3
45	208.4	208.8	209.9	6.00	183.1	184.5	187.5
46	213.0	213.4	214.5	6.13	187.1	188.5	191.7
47	217.6	218.0	219.1	6.26	191.2	192.6	195.8
48	222.2	222.6	223.8	6.39	195.2	196.6	200.0
49	226.8	227.2	228.4	6.52	199.2	200.7	204.2
50	231.4	231.8	233.0	6.65	203.3	204.7	208.3
55	254.4	254.9	256.2	7.30	223.3	225.0	229.1
60	277.4	277.9	279.3	7.95	243.5	245.3	250.0
65	300.4	301.0	302.5	8.58	263.6	265.7	270.8
70	323.4	324.0	325.7	9.24	283.9	286.0	291.7
>70	SUS = 4.62 × cSt	SUS = 4.629 × cSt	SUS = 4.652 × cSt	°E = 0.132 × cSt	4.049 × cSt	4.082 × cSt	4.167 × cSt

The most common unit of measurement of kinematic coherence is cSt (centistokes), where 1 St = 100cSt = 1.0 cm2s-1. However, other units like Engler degrees, seconds of Redwood and S.U.S. (Saybolt Universal Seconds). Table 2.1 can be used for changes of kinematic coherence values between the previous units for three specific temperatures.

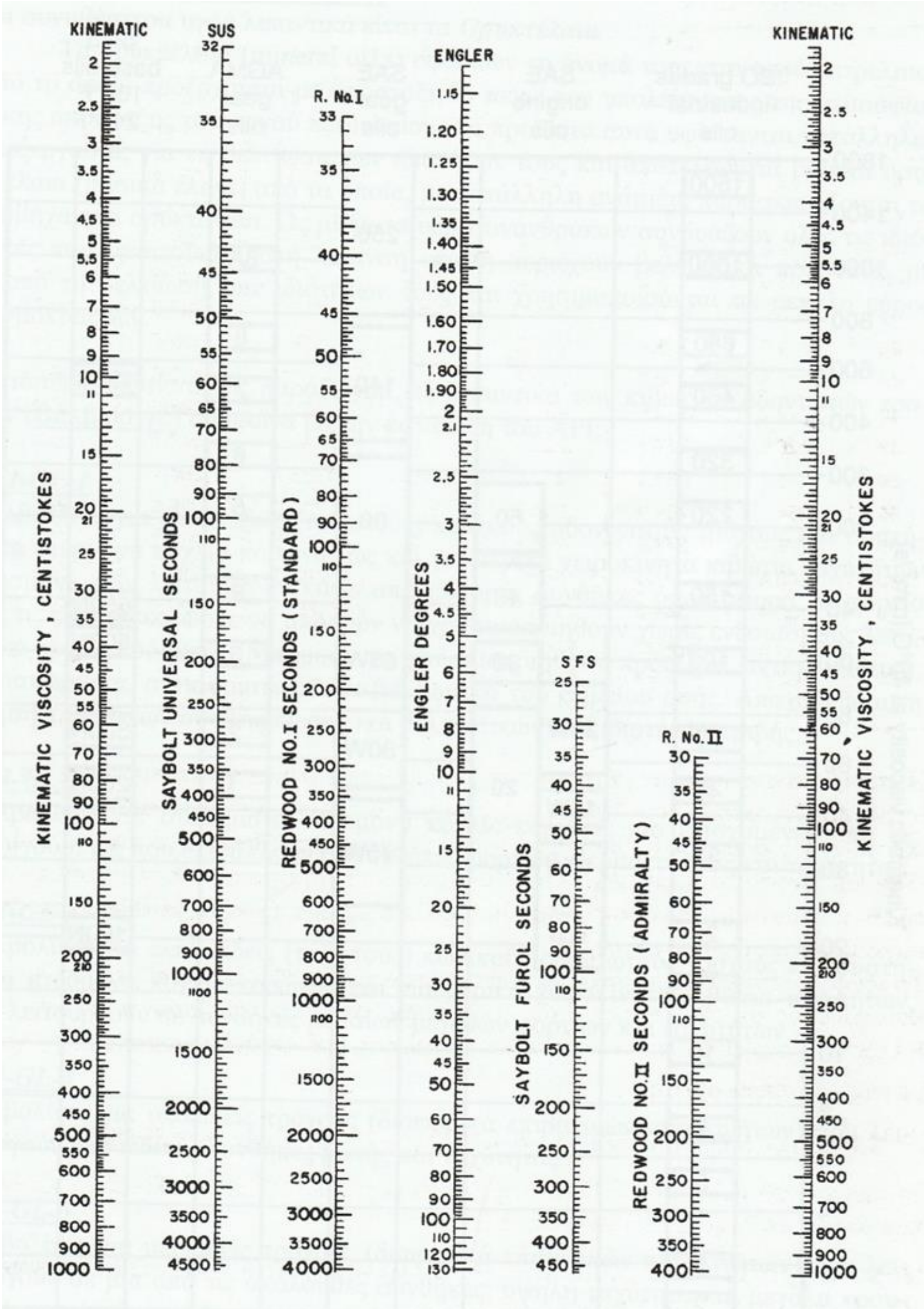
Viscosity is converted only at the same temperatures. Conversion of the kinematic viscosity to a temperature t from cSt to S.U.S. (that's the finding of S.U.S. (t) when viscosity is known at cSt on given temperature t) is done with the following relation:

S.U.S (t) = S.U.S (100°F) • [1 + 6.4×10-5(t-100)] where S.U.S. (100°F) is the kinematic viscosity in S.U.S. at the temperature of 100°F of the given viscosity which can be taken from table 2.1.

Example 2.1 how many S.U.S. are at 30 cSt at 210°F; for 30 cSt at 100°F we take from table 2.1, 140.9 SUS,

Consequently,

S.U.S. (210°F) = 140.9[1 + 6, 4×10-5(210-100)] = 141.9 S.U.S.



Sketch 2.6 Viscosity conversion code in various units.

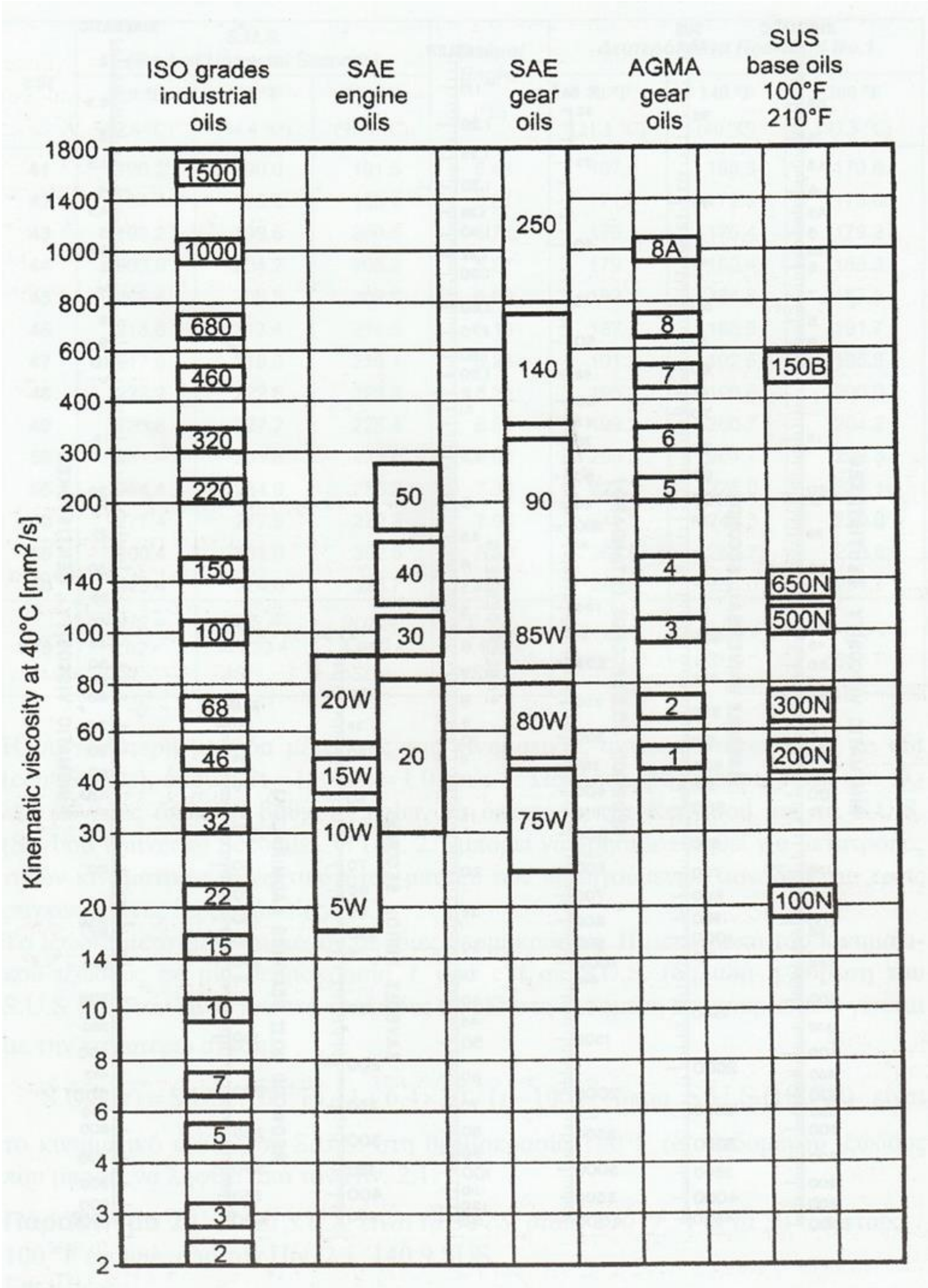


Table 2.2 Comparison table of various lubricants.

2.2.2 Liquid lubricants (oils)

The most common liquid lubricants are mineral oils.

Mineral oils owe their name to crude oil, from which they come by vacuum distillation of the residual atmospheric distillation of crude oil. These products undergo appropriate treatments to improve their properties and constitute the basic mineral oils (essential oils) from which, with appropriate mixing, the industrial mineral oils are prepared. As mixtures of hydrocarbons they combine all the properties required in lubrication because they contain improving additives, in order to improve their properties and are used in a wide range of temperatures.

The following are examples of gearbox lubricants (Valvoline) according to the classification of API:

API-GL-1

Valvolines for helical (arched) conical gears and endless screw-crown systems as well as for some manual transmissions (chassis), which operate under light conditions in terms of loads and speeds so that they can be used as much as possible. Additional antioxidants, anticorrosive, antifoams and flow point improvers may be used. The use of high pressure additives and abrasive properties is prohibited.

API-GL-2

Valvolines for endless screw-crown systems more reinforced than the previous ones imposed by the conditions of load temperature and speed.

API-GL-3

Valvolines for helical (arched) conical gears and endless screw-crown systems and for some manual gearboxes, operating in medium-sized load and gear conditions.

API-GL-4

Valvolines for sub-wheels (differentials of passenger cars) operating under torque and speed conditions.

API-GL-5

Valvolines for sub-wheels (differentials of passenger cars) operating in one of the following conditions: high speed and low torque or low speed and high torque. This specification together with API-GL-4 covers most of the cases of lubrication of cogwheels in cars.

Note:

Water may be present in the lubricant either as a separate layer or in dilution. The presence of water in the lubricant is undesirable because it reduces the lubrication and contributes to oxidation. Water has a low capacity for lubrication and a big problem in its use because it is very corrosive, but it has a great cooling capacity.

2.2.3 Semi-solid or semiliquid lubricants (fats or greases)

Grease is a semi-solid product of a thickener in a liquid lubricant. This product may contain ingredients that give it special properties.

They are animal and vegetable fats and oils and saponified fats. During the operation phase of the machines in which they are used, they are liquefied due to the increased friction temperature. They have all the properties of liquids and an excellent lubricating capacity due to low coefficient of friction. They are stable because they are easily oxidized with the result that their long-term use is detrimental to the lubrication position and therefore it is necessary to constantly renew the lubricants in the repositioning positions.

The most common lubricant fats (greases) are those delivered from mineral oils, which under the influence of solidifying substances (thickeners) are converted into a mass of semi-solid or semi-liquid state. Superior fatty acid soaps, inorganic substances and polymers are commonly used for their preparation. Most lubricating fats are produced from soaps of lithium, calcium, sodium, aluminum or complexes.

<u>The role of grease</u>

For each specific application the role and function of the grease is:

1. To lubricate so as to offer protection against wear.

The factors that regulate this property are: the lubrication capacity of this type of grease, which is a function of the coefficient of friction and the resistance of the grease to deformation and mechanical shear.

2. To continue to the sealing of the system.
3. To protect the surfaces from the influence of the external environment (humidity, corrosive vapors, dust, etc.)
4. To show resistance to thermal phenomena as well as to the stresses resulting from the operating conditions (local overloads, vibrations, abrupt changes of speed, etc.)
5. To maintain for as long as possible its original properties, which presupposes mechanical stability (especially important for bearing applications).
6. Not to oxidize.
7. To show resistance to hydrolysis and not to change its basic characteristics with the presence of limited amounts of water.

8. Be compatible with elastomers and construction materials of the mechanism.
9. Not to increase its consistency too much at low temperatures, so as not to show great resistance to movement.
10. In its storage and transport conditions that its original properties do not change.

In special cases the greases must have other additional characteristics, such as not detaching from their place of application (textiles, plastics, food, etc.), yet, not dissolving from petroleum products (pumps in petroleum and chemical plants), radiation (nuclear plants). Especially for aviation applications we can say that special greases are needed both due to the range of operating temperatures and for the relatively limited quantities that due to the limitation of the weight and volume of the aircraft must be consumed in them.

Grease advantages

The only way to obtain appropriate properties the greases require for special applications, is to prepare them on the basis of synthetic oils. The main advantages of greases over other lubricants are:

1. In applications where the design of the system cannot prevent the leakage of lubricant (which can contaminate the products produced or create a risk of ignition) the use of grease is recommended.
2. These are cases where high operating temperatures preclude the use of common, conventional oils.
3. Synthetic oils are used in some special applications, but most of the time the solution is given by greases.
4. Grease is used in bearings. Mainly in those that are in a place with difficult access or the system operates for long periods of time without special maintenance.
5. When the speed of movement is very low and the load very high and oil is used, there is a case that a lubricating layer does not form.
6. In applications such as construction or earthmoving machinery, where there is a lot of dust in the atmosphere, the lubricant must seal the system to prevent dust from entering its interior. In these cases the grease provides very good protection, as in applications where protection against rust from moisture in the atmosphere is needed.
7. Lubrication systems and devices are much simpler and cheaper than their oil counterparts.

Disadvantages of greases

The most important disadvantages of grease that bother manufacturers are:

1. Because the flow capacity and circulation speed of the grease is very small in relation to oil, there is not the same possibility of heat dissipation from the system.
2. In several greases the fact of the change of their structure with time is observed. In some the oil separates and hardens and others become softer with the effect of vibration, moisture, storage, etc.

2.2.4. Lubricant additives

Crude oil distillation products have proven to be satisfactory lubricants for various types of engines for several years. Today most modern machines require the use of more complex lubricants which are, with addition of a wide variety of ingredients known as additives.

Additives are needed in lubricants for the following reasons:

1. Improve one or more of the original properties of the base oil and give the final product new characteristics.
2. Reduce the rate of quality degradation of their properties (anti-aging) with the passage of time of operation of the machine.

Many of the categories of additives while performing more than one function are nevertheless characterized by their main action. They are commercially available as mixtures containing a range of several additive categories. We achieve different levels of performance depending on the required specifications by using them in different percentages.

The choice of additive (or combination of several) to achieve a specific specification must be made carefully, as some additives react with each other or with other components of the lubricant and produce adverse reactions, or sometimes the effect of cooperating with each other is observed and to give the lubricant characteristics, which exceed the original specifications. It is obvious that the composition of a lubricant for the satisfaction of a specific project is not just a mixture of additives with an essential oil but a combination of many factors.

<u>High pressure additives and wear protection</u>

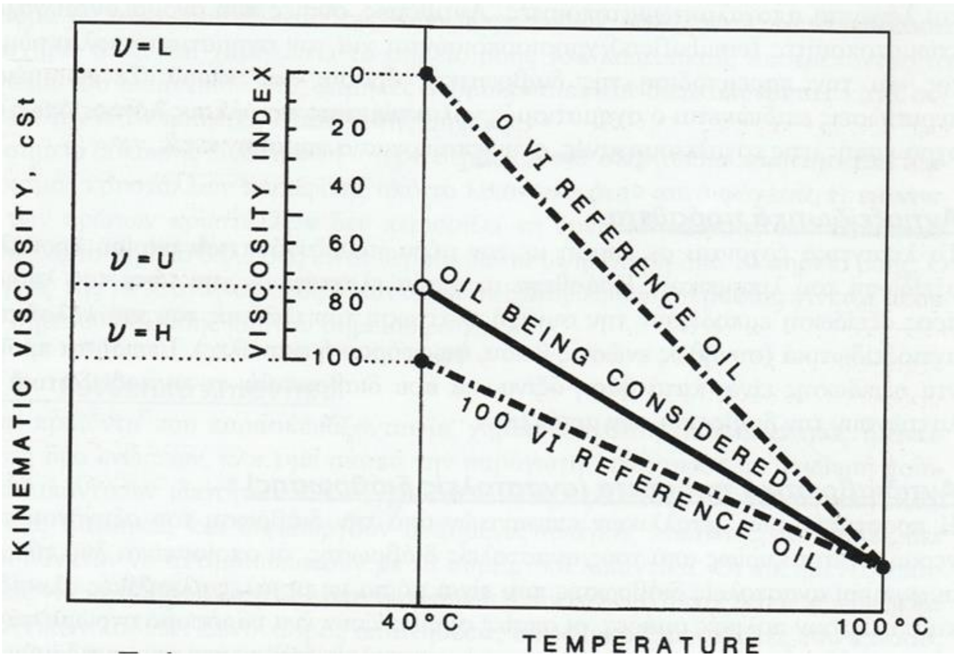
One of the most important functions of a lubricant is to reduce the protection of metal chips, such as two metal surfaces friction products, and to prevent cracks and scratches as they prevent metal contact. The additives that give the lubricant this property in combination with the high pressure additives react with the metal surfaces and create at high temperatures a thin but strong layer which does not allow the contact of metal with metal, as in the lubrication of indentations.

Friction adjusters

In combination with the high pressure additives the friction regulators are used, which create a thin lubricating layer on the frictional metal surfaces. Thus reducing static friction at start-up and thus reducing energy consumption and wear. They are usually fatty acids and their derivatives or various types of waxes (e.g. fatty amines).

Viscosity index enhancers

The measure of the resistance of a lubricant to changing its viscosity is called the viscosity index. The addition of viscosity enhancers, in the form of polymer additives, helps to reduce the change in viscosity as a function of temperature. These additives improve the lubrication characteristics of the lubricants, increasing the viscosity index and achieving stable lubrication with the change of temperature.



Way of finding VI

In lubrication applications where temperatures remain relatively constant, the viscosity index is of little importance. However, when the operating temperature varies a lot, as in car engines, the viscosity index of the oil to be used must be as high as possible, always in accordance with the other properties of the oil.

Additives against foam production

Foam is produced from the air that is trapped in the lubricant. If there is foam then an amount of lubricant is reduced and the lubrication is defective.

The presence of additives that are against the production of foam prevent the entry of air bubbles into the lubricant by reducing the surface tension, achieving the destruction of the bubbles. The antifoaming additives used are actually high molecular weight silicones with a high surface tension that cause the bubbles to coalesce into larger ones that rise to the surface and are destroyed.

<u>Emulsifiers</u>

During the operation of the machines many lubricants come in contact with water, from which they must be able to be easily separated. For this separation are used additives that prevent the formation of emulsions and are called emulsifiers. Instead, substances called emulsifier are used to form emulsions and inhibit the corrosive action of water. In some cases the formation of emulsion is sought for other reasons such as cutting fluid in machine tools, thread processing, etc.

<u>Antioxidant additives</u>

The lubricants come in contact with the air, oxygen of which causes the lubricant to oxidize. Additives that reduce the tendency of the oil to oxidize prevent a continuous decrease in viscosity and are called antioxidants (usually sulfur, phosphorus or phenol compounds). Because oxidation products are essentially acidic and therefore corrosive, antioxidants reduce the corrosion of metals.

<u>Anti-corrosion additives (corrosion inhibitors)</u>

The protection of metal surfaces from corrosion of water oxygen is mainly done by corrosion of the inhibitors, which are of two types:

Natural corrosion inhibitors which are molecules with large alkyl chains and contain polar groups, which form a hydrophobic layer on the metal surfaces and chemical corrosion inhibitors that react with the metals and form strong electrode layers which change the electromechanical potential of the surfaces. These include fatty acids that are effective only in the presence of water or other polar compounds. Corrosion and rust inhibitors adhere to the lubricating surface and prevent corrosive products of the lubricant from coming into direct contact with it.

<u>Cleaning additives</u>

Lubricating oils exposed to high temperatures and escaping high consumption temperatures gases tend to produce highly acidic components which in turn polymerize and synthesize materials with low solubility in the oil. Detergent additives prevent the deposition of such products by neutralizing acidic ingredients. As such additives are used mainly compounds of barium, magnesium and calcium, which are also excellent rust suppressors.

Flow properties at low temperatures

The flow point gives the temperature at which paraffin separation is so intense that it does not allow the lubricant to be fluid. Most base oils (basic mineral oils) contain small amounts of paraffin, which, however are capable of forming crystals at low temperatures that impede the smooth flow of a lubricant. A suitable additive reacts with the paraffin crystals, mainly by changing their size and shape and lowering the flow point of the lubricant, preventing it from freezing at low temperatures and thus facilitating the satisfactory lubrication of the machine.

The point of turbidity is the highest temperature at which separation of paraffin crystals from the lubricant is observed when it cools. The appearance of the first crystals does not limit the flow potential of the fuel. The blur point is usually temperatures higher than the flow point. The control of the fluidity of a lubricant at a low temperature is done through the turbidity point and the flow point.

2.2.5. Synthetic lubricants

They are the products that are prepared with chemical composition (polymerization or reaction of two compounds, etc.) with the aim of producing lubricants with constant quality lubricating properties.

Modern engines operate under very severe conditions and create increased lubrication needs, which cannot be met with conventional lubricants. Increased demands for high quality lubricants have contributed to the development of synthetic lubricating oils. A combination of requirements such as small changes in viscosity and good lubrication properties over a wide range of temperatures, chemical stability at high pressures, good behavior at low temperatures, good characteristics against corrosion and foam production, i.e. they can be produced very specially.

Many classes of organic compounds have been used as essential oils in the past. Indicatively some of them are polyether, esters, phosphoric acid esters, silicon and synthetic lubricants. It is also possible to mix a synthetic lubricant and a mineral oil. The final product is characterized as a synthetic lubricant if its mineral oil content is less than 14%. But synthetic lubricants do not always give the best expected solutions and do not have only advantages. A major drawback is that synthetic oils are very expensive compared to mineral oils.

2.2.6. The recycling of lubricants

A field of processing mineral oils, which has experienced great technological development internationally, and which in recent years has developed in Greece, is the recycling of lubricants. Used lubricants pose a risk to the environment, because they contain a significant number of chemicals that are harmful and can cause ecological damage. It is indicated that a liter of lubricant dumped on a water surface can cover an area of 1000 square meters and thin surface layer that is created is enough to catastrophically disrupt the various forms of aquatic life beneath it.

Today, a large part of the used lubricants are either disposed of directly in the environment or collected and used as fuel, polluting the environment (soil and groundwater) with toxic metals chemical pollutants. However, most of the waste oil used is recyclable and can be refined, producing essential oils of the same or better quality than the original essential oils used to produce industrial oils and the pollutants can be safely separated and removed.

2.3 TECHNICAL ELEMENTS OF INDUSTRIAL GREASES

Even the best machine can work well only if it has proper lubrication. It is especially important to choose the right lubricant-grease and the right way of lubrication as well as the re-greasing interval. For this reason, manufactures' researchers and engine maintenance technicians consider lubrication to be one of the most important issues in bearing applications, as important as the choice and construction of the bearing itself. The experience in bearings was the basis for the creation of high quality lubricants that evolve through continuous research and testing. Each lubricant is produced to fit the type of application for which it is projected.

Grease selection for bearings

Improper lubrication is the cause for 36% of cases of premature bearing wear. General purpose greases are insufficient to meet the special needs of special bearings. In bearing applications one encounters a wide range of different operating conditions and therefore the lubrication should exactly match the type of application.

Bearing grease provides smooth, hassle-free and highly reliable operation under the most adverse conditions. It reduces the entry of foreign particles into the bearing, reduces impact loads and protect against rust, etc.

Choosing the right lubricant for an application is essential to achieve maximum machine life.

The basic criteria for the selection of a lubricant are the type and size of the bearing, the speed and loads, as well as the duration of operation and the re-greasing time.

The operating temperatures of a grease depend mainly on the type of base oil, the soap and the additives. The typical operating temperatures of a grease are:

LTL Low-temperature limit: is the lowest temperature that the grease allows the bearing to start.

LTPL Low-temperature performance limit: below this temperature, lubrication is deficient on the contact surfaces of the rolling elements with the grooves. Prices are different for Ball bearing and Roller bearings.

HTPL High-temperature performance limit: above this temperature grease oxidizes and so its lifespan cannot be calculated.

HTL High-temperature limit: when this temperature is exceeded the structure of the grease is destroyed (e.g. the flow point).

<u>Oil separation</u>

Greases release liquid lubricant with the increase of temperature or when stored for long periods of time. The degree of separation of the liquid lubricant depends on the thickener, the base oil and the production method. To measure this loss of grease we fill the container with weighed amount of grease and place a weigh of 100 gr on the free surface of the grease. The whole system is placed in an oven at 40°C for a week. At the end of the week, the amount of liquid produced is weighed and reported as a percentage of fat loss by weight.

<u>Lubrication capacity</u>

The lubricating capacity of the grease and its efficiency at high temperatures are confirmed by simulating with a special device the operating conditions of the large ball bearings in bearings. The tests are performed under two different tests. The first test A is performed at room temperature and the second test B at temperatures of 120°C. Success in test A means that the grease can be used to lubricate large bearings at normal operating temperatures and also in low vibration applications. Success in test B means that the grease is suitable for use in large bearings at high temperatures.

<u>Grease life for bearings</u>

Special grease testing machine determines its lifespan and performance at high temperatures. Ten monosphere bearings are mounted on bearings and filled with a given amount of grease. The test is performed at a predetermined speed and temperature. Axial and radial loads are exerted simultaneously and the bearings operate until they are damaged. The time is recorded in hours and calculated for the lifespan, according to Weibull, at the end of the tests, to find the lifespan of the grease. This information can be used to determine the bearing time of bearings.

Redox interval

Choosing the right grease for an application is very important for bearing performance. Equally important is the addition of the right amount of grease at the correct intervals depending on the type of bearing, the application and the properties of the grease selected. Excessive or incomplete lubrication as well as the use of unsuitable re-greasing methods reduce the service life of the bearing.

Lubrication methods

The method of lubrication is as important as the choice of grease and re-greasing. The use of lubricants, manual or automatic, facilitates the proper lubrication of the application. However, lubrication must be done in a clear environment to avoid the entry of foreign particles that can damage the bearing. The use of grease meters in combination with grease guns and pumps ensures the supply of the right amount of grease. Continuous lubrication with the use of automatic lubricants, single or multiple points, enables stable and controlled lubrication, thus reducing the risk of excessive or insufficient lubrication and helping to maximize bearing performance. In addition, automatic re-greasing reduces the risk of contamination.

Examples of recognizing the symbols of a grease

Let the grease K P 2 G – 20 according to DIN 51825

K = grease bearings (G = grease for closed gears, OG = grease for open gears, M = friction gears for bearings/seals)

P = additional information (P = EP additives, F = solid lubricant, E = esters)

2 = NLGI grade (see NLGI classification)

G = maximum operating temperature and moisture resistance (see table of maximum operating temperature classification)

-20 = minimum operating temperature

Letter designation for maximum operating temperature t_{max}

Letter	Maximum Temperature t_{max} (°C)	Letter	Maximum Temperature t_{max} (°C)
C	+60	M	+120
D	+60	N	+140
E	+80	P	+160
F	+80	R	+180
G	+100	S	+200
H	+100	T	+220
K	+120	U	>+220

Maintenance of layer and form of a grease

Hardness of grease is classified according to the scale that was developed by NLGL (National Lubricating Grease Institute). That is based on the degree of penetration of a standard cone that is allowed to sink in a temperature grease of 25°C for 5 seconds. The depth of penetration is measured in the scale of 1/10 of the millimeter (10-1 mm). The softer greases allow the cone to sink deeper, so they have a greater penetration degree.

Classification of greases according to NLGI scale in relation with the sustainability of the layer and form of the lubricant as a function of the degree of penetration according to ASTM.

NLGI grade	ASTM (10^{-1}mm)	Form at normal temperature	NGLI grade	ASTM (10^{-1}mm)	Form at normal temperature
000	445-475	Very fluid	3	220-250	Medially hard
00	400-430	Fluid	4	175-205	Hard
0	355-385	Semi-fluid	5	130-160	Very hard
1	310-340	Very soft	6	85-115	Extremely hard
2	265-295	Soft	-	-	-

Bearing operation parameters

Temperature

Symbol	Temperature limits	Temperature °C
L	Low	<50
M	Medium	50 to 100
H	High	>100
EH	Extremely high	>150

Ball bearings' speed

Symbol	Temperature limits	$n \cdot d_m$ (rpm·mm)
L	Low	<100 000
M	Medium	Up to 300 000
H	High	Up to 500 000
VH	Very high	Up to 700 000
EH	Extremely high	>700 000

Load

symbol	Speed limit	C/P
L	Low	15
M	Medium	8
H	High	4
VH	Very high	<2

<u>General grease selection board LG – lubricating grease</u>

General purpose grease = LGMT - 2

It is considered general use when speed = M, temperature= M and load = M

(For areas with relatively high ambient temperature grease LGMT – 3 is better).

High temperature grease = LGHP - 2

When the bearing temperature is expected to be constantly > 100°C

Extremely high temperature grease = LGET - 2

When the bearing temperature is expected to be constantly > 150°C

Low temperature grease = LGLT - 2

When the bearing temperature is expected to be < 50°C and the environment's -50°C

Grease for high load = LGEP - 2

Axial and large loads with frequent starts and stops.

Grease for food related process = LGFP - 2

Grease with low toxicity requirements = LGGB – 2

Green, ecological and biodegradable.

2.4 INDUSTRIAL GREASES AND APPLICATIONS

<u>LGMT - 2</u>

It is a general purpose grease for industry and the automobile.

The LGMT – 2 is mineral oil grease with lithium soap.

It is for general purpose and of excellent quality and it's suitable for a wide range of applications in industry and automotive.

It has very good oxidation stability, good mechanical stability and good anti-rust properties and water resistance.

It is suitable for medium temperatures, low to medium speeds and high loads even with vibrations.

Typical applications: agricultural tools. Small electrical appliances. Car wheel bearings. Conveyor belts. Industrial fans.

<u>LGET - 2</u>

Grease of extremely high temperature and extreme conditions. LGET – 2 is a high quality grease, based on synthetic fluorescent oil and a special thickener.

It has excellent lubrication properties at very high temperatures above 200°C up to 260°C.

It has a long life in an environment with many pollutants, especially in an environment with a potential for explosion or in areas with the presence of very pure gases of oxygen, hexane, etc.

It has high resistance to oxidation and corrosion and excellent resistance to water and steam.

Typical applications: bakery tools (ovens). Blast furnace wheels. Biscuit baking machines. Textile dryers. Load rollers on printing machines. PVC packaging machines. Electric motors at extreme temperatures. Exhaust fumes. Vacuum pumps.

LGMT - 3

General purpose grease for industry and the automobile. LGMT – 3 is mineral oil grease with lithium soap.

It is general purpose and of excellent quality and it's suitable for a wide range of applications in industry and automotive.

Has excellent anti-rust properties.

Has high resistance to oxidation in the range of its operating temperatures.

It is suitable for medium temperatures and speeds for low to medium loads.

Typical applications: bearings with outer ring rotation. Applications on a vertical axis. Bearing on shaft over 100 mm continuous high ambient temperature above 35°C. Axles are propellers. Agricultural tools. Car and truck wheel bearings. Large electric motors.

LGEP - 2

Grease for high loads and high pressures. LGEP – 2 is a high quality grease based on mineral oil and lithium soap and contains additives for high pressures. This grease provides very good lubrication at medium operating temperatures from -20°C to 110°C. It has high mechanical and antioxidant stability and high performance at high pressures. It is suitable for low to medium speeds and for large loads even with intense vibrations.

Standard applications: paper machines. Crushers. Train engines. Dams. Roller work for roller. Heavy machines. Sieves. Crane wheels.

LGFP – 2

Grease bearing compatible with food. LGFP – 2 is pure non-toxic grease, without hydrocarbons, does not stain, is based on white medical oil and uses aluminum alloys as soap.

Is formed using only permitted ingredients per FDA (Food and Drag Administration) and has been certified by NSF, for maintenance category H1 (incidental contact with food).

It is compatible with all food regulations.

Has a high resistance to water which makes it suitable for applications where washing is done very often.

Has features that reduce bearing wear.

Has high resistance to oxidation. Has a neutral value ph.

Typical machines: bakery and food processing tools. Food packaging machine bearings. Wrapping machines. Bearings in conveyor belts. Bottling machines.

<u>LGEM – 2</u>

Grease for bearings. It has high viscosity and solid lubricants.

LGEM - 2 is based on mineral oil and thickener a lithium soap containing molybdenum disulfide and graphite.

It shows good lubrication in conditions of large loads and low speeds.

It is suitable for medium temperatures and low speeds as well as for high very or very high loads even with intense vibrations or shocks.

Typical application: bearings that move at low speeds and very large loads. Crushers. Truck. Transport. Lifting wheels. Construction machinery such as mechanical arms, crane arms and crane hooks.

<u>LGEV – 2</u>

Grease for bearings. It has very high viscosity and solid lubricants. The LGEV – 2 is a high quality grease, very high viscosity, based on mineral oil and lithium-calcium soap that contains molybdenum disulfide and graphite.

It has excellent lubricating properties because it contains molybdenum disulfide and graphite. It is excellent for application in large barrel bearings with high load and low speed and where slipping can occur.

It has high mechanical stability and resistance to water and corrosion.

It is suitable for medium temperatures and for loads with vibrations or shocks and oscillating movements.

Standard applications: bearings for rotating drums. Supports rotating blast furnaces and dryers. Pockets. Crown bearing. Rolling rollers. Crushers.

LGLT – 2

Grease for low temperatures and very high speed. LGLT – 2 is a high quality grease, based on synthetic oil and lithium soap. The special properties of thickener and the low viscosity of the oil provide very good lubrication at low temperatures and at very high speeds. It is suitable for low friction torque, low energy losses, quiet operation, has high oxidation stability and water resistance.

Typical applications: textile machine heads. Machine tool heads. Control instruments and tools. Small electric machines for medical and dental instruments. Discount machine cylinders. Robots.

LGGB – 2

A green (ecological) biodegradable grease for bearings. LGGB – 2 is a biodegradable, low toxicity grease, with lithium-calcium soap.

It has excellent lubricating properties for a wide range of applications in different operating conditions.

It is suitable for low to medium temperatures and speeds and for medium to high loads even with vibrations or shocks.

It is compatible with the regulations on toxicity and biodegradation.

It has good behavior in metal by metal cooperation applications, such as ball bearings, ball bearings and cylindrical bearings.

It has good behavior at low starting temperatures and good anti-corrosion properties.

Typical applications: agricultural and forestry machinery. Construction and digging machines. Mining and transport tools. Water treatment and irrigation. Dams. Bridges. Links and clamps. Applications where environmental pollution is taken into account.

LGWM – 1

Grease for very high pressure and low temperatures. The LGWM – 1 is based on mineral oil with lithium soap and contains high pressure additives. It is extremely suitable for lubrication of bearings that operate with a good combination of radial and axial loads (e.g. transport screws).

It has a good lubrication membrane formation at low temperatures (up to -30°C).

It has good compressibility in piping at low temperatures.

Has good corrosion protection, and good water resistance.

It is suitable for low to medium speeds and for high loads even with impacts.

Typical applications: windmills. Screw conveyors. Central lubrication systems. Applications with thrust barrel bearings.

LGWA – 2

Grease for wide range of temperatures for high pressure and load. LGWA – 2 is a high quality grease, based on mineral oil and lithium soap.

It has very good properties and can be installed in a wide range of applications in industry and vehicles. It shows excellent lubrication potential for temperatures up to 220°C (but for a short time).

It protects the wheel bearings of vehicles under adverse operating conditions.

It has efficient lubrication in wet conditions.

Good resistance to water and corrosion.

Shows very good lubrication from high loads and low speeds.

Typical applications: car and wheel bearings. Washing machines. Electric motors.

LGHB – 2

Grease with high viscosity for high temperatures and high load.

LGHB – 2 is a high quality grease based on mineral oil that uses calcium sulfide soap. It does not contain additives and the high pressure characteristics are created by the structure of the soap.

It has very good properties against oxidation and corrosion. Very good behavior at high pressure in applications with high loads.

Typical applications: paper machines. Roller work for rollers. Forklift lifting mechanism. Continuous casting machines. Road construction machinery. Closed barrel bearings with a capacity of up to 150°C. Withstands temperatures of 200°C for a short time.

<u>LGHP – 2</u>

High efficiency and high temperature grease. LGHP – 2 is a high quality grease based on mineral oil that uses as a thickening di-urea.

It is suitable for bearing applications with silent operation and with wide temperature range -40° C to 150°C.

It is suitable for medium and high speeds and for low to loads.

It has a long service life at high temperatures and a large temperature range with high thermal stability.

Good performance at low degrees. Very good corrosion protection.

It is compatible with polyuria grease, with grease having a thick lithium complex.

It has silent operation with very good mechanical stability.

Typical applications: textile bearings, paper machines and dryers. Electric machines. Industrial fans. Water pumps. Ball bearing applications at high speeds and medium and high temperatures. Clutch bearing. Oven wagons. Applications with vertical axis. Vibration applications.

2.5 WEAR OF MATERIALS

Lubrication analysis is the technique that examines different lubrication (oils, greases, etc.) in terms of their ability to provide acceptable quality lubrication to the mechanical equipment, in order to avoid wear.

Some types of lubricant analysis techniques provide a fairly accurate estimate of the degradation of the various chemical components of the lubricants in terms of their original properties. A comparison of the amount of metal particles that resulted from the wear of the surfaces, during the analysis of samples taken periodically, can reveal the type of wear of the lubricated parts of the machines and the extent to which these wears have taken place.

Lubrication analysis is an important aid to preventive maintenance. However, it is limited to maintaining the lubricants at an optimal level of operation without being able to accurately determine the critical production equipment needs for preventive maintenance. The use of lubrication analysis as a technique of prognostic maintenance exceeds its potential. Claiming that it can replace the method of vibration analysis leads to the misconception that it is a tool of predictive maintenance.

Lubricating analysis can be limited to applications where premature wear of large amounts of lubricant has become widespread. The methodology can be applied to detect the degradation of the properties of the lubricants and allow the user to extend the life of the lubricant, by refining it from unwanted particles, separating the water or replacing the additives with new ones.

With few exceptions, the analysis of lubricants is done in laboratories as the purchase cost of the analysis systems and staff training for their use within the production facilities is prohibitive. During the analysis of the condition of the lubricant, the parameters that represent the properties of the lubricants are examined, according to which the suitability of the lubricant to create the necessary lubricant membrane between the relatively moving surfaces is judged.

The classical method of lubricant analysis cannot determine the true cause of the lubricant deterioration. For this reason the other analysis techniques offered must be used in a prognostic maintenance program to find the cause of which the lubricant has deteriorated.

Lubricant spectrum analysis allows fast and accurate measurements of many of the components that make up the lubricant. These particles are classified as perishable metals, contaminants and additives. Some items may belong to more than one of the above categories. This technique measures molecular compounds such as water, refrigerants as well as liquid fuels in the lubricant.

<u>Wear analysis</u>

The analysis of wear due to foreign particles in the lubricant is not related to the analysis of the lubricant, except that the particles to be studied are collected through a sample of lubricant. This technique provides us with direct information about the evolution of damage inside the machine. The shape of the particles, their composition, their size and their content in lubricant are carefully studied for this purpose. Particle analysis can be applied by one of the following two methods:

First method

It concerns the routine analysis and study of particle size, shape, and content. For normal operation the diameter of the particles must not exceed 10μm and be at a low content in the mass of the lubricant. In general, the combination of the three techniques described below is used to collect and analyze the particles that cause decay:

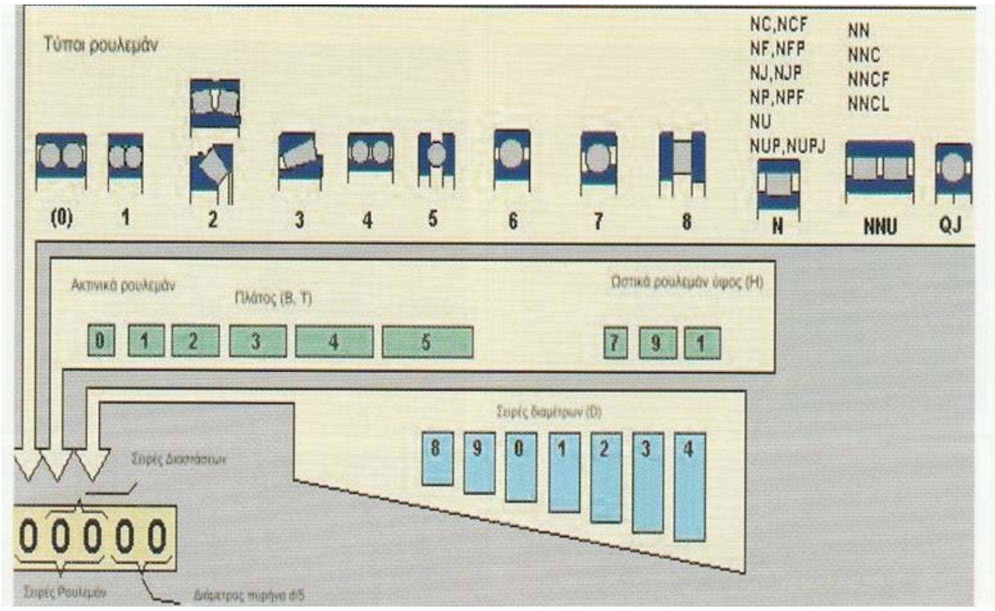
Direct control of the main circulating lubrication system during operation: devices such as magnets, coils or different pressure gauges are installed in the lubricant circulation network to control the main flow of lubricant inside the machine. Differential pressure gauges located in the filter positions can be used to control number of particles discharged from the circuit. The filters can collect all the particles, magnetic and non-magnetic, but the size of the particles they hold depends on the size of the filter pores.

Control through secondary branch for analysis during operation: this method uses a secondary branch of the main circuit from which it is possible to measure the turbidity of the lubricant by means of optical devices, in which case corresponding conclusions are drawn for the concentration of unwanted particles in the lubricant. Like the previous method, it cannot offer an analysis of the real causes of lubrication wear.

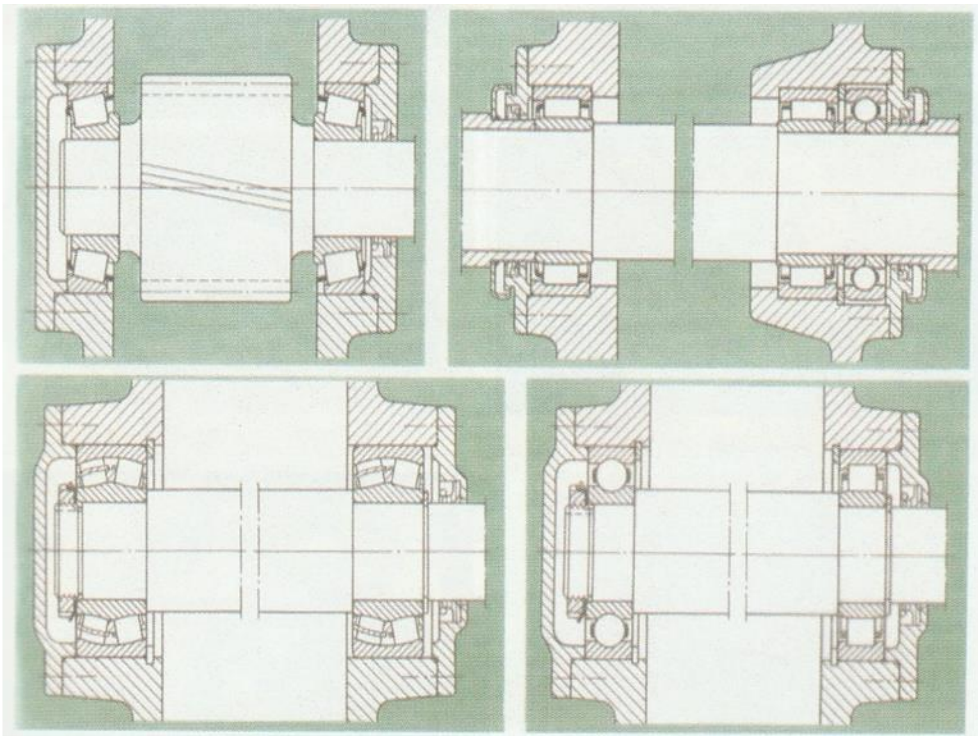
Lubricating test in the laboratory: the only method that can identify and fully analyze the causes of degradation of the lubricant is the analysis in the laboratory, i.e., out of the production.

Second method

The second method concerns the investigation of the material of solid particles. The five main types of wear mentioned can be noted depending on the type of particles: (1) wear due to friction, (2) wear due to removal of the surface layer, (3) wear due to fatigue from rolling, (4) combination due to rolling and slipping and (5) intense wear. Particles that are smaller than 15μm are produced only upon wear of formulas (1) and (3). More about damage is mentioned in Chapter 4.

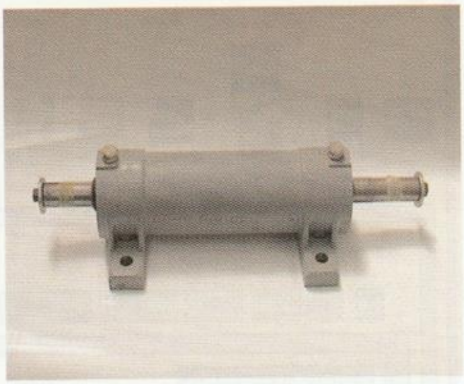


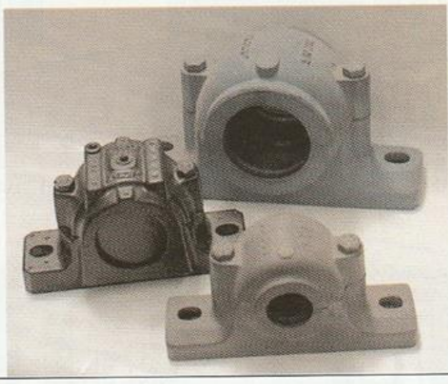
Symbolism and types of rolling bearings



Shaft support with rolling bearings


Μοντάρεται με 6 απλές κινήσεις αντί 28 που απαιτούνται για το συμβατικό.
Ξεμοντάρεται σε ένα λεπτό.



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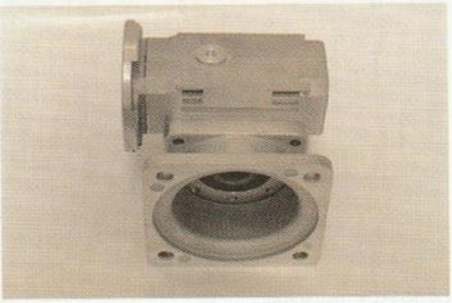
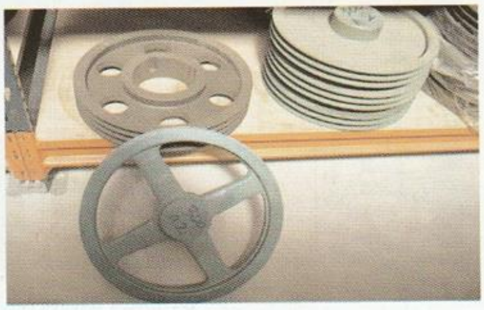
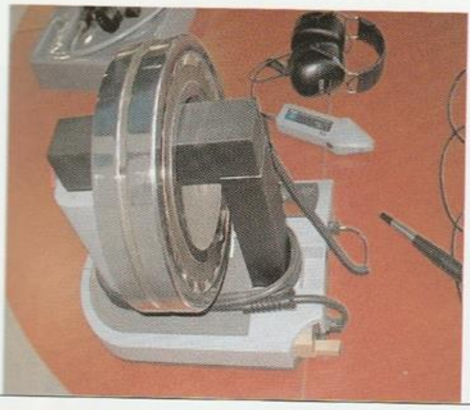
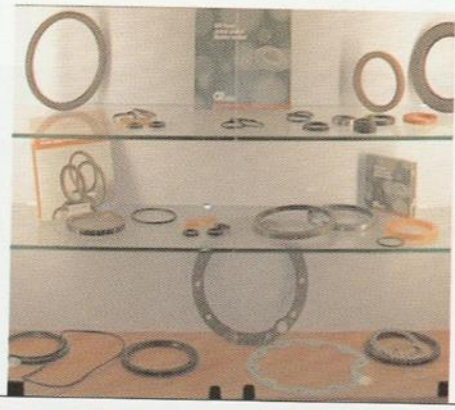
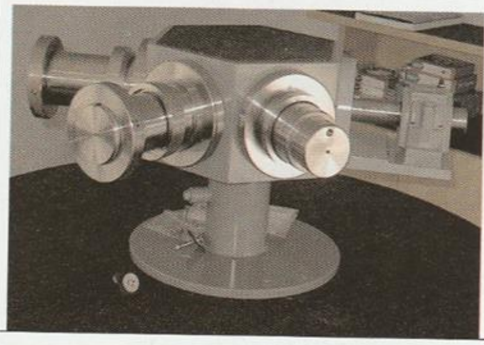


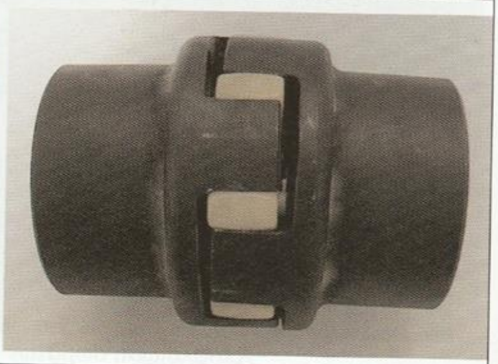







SPM



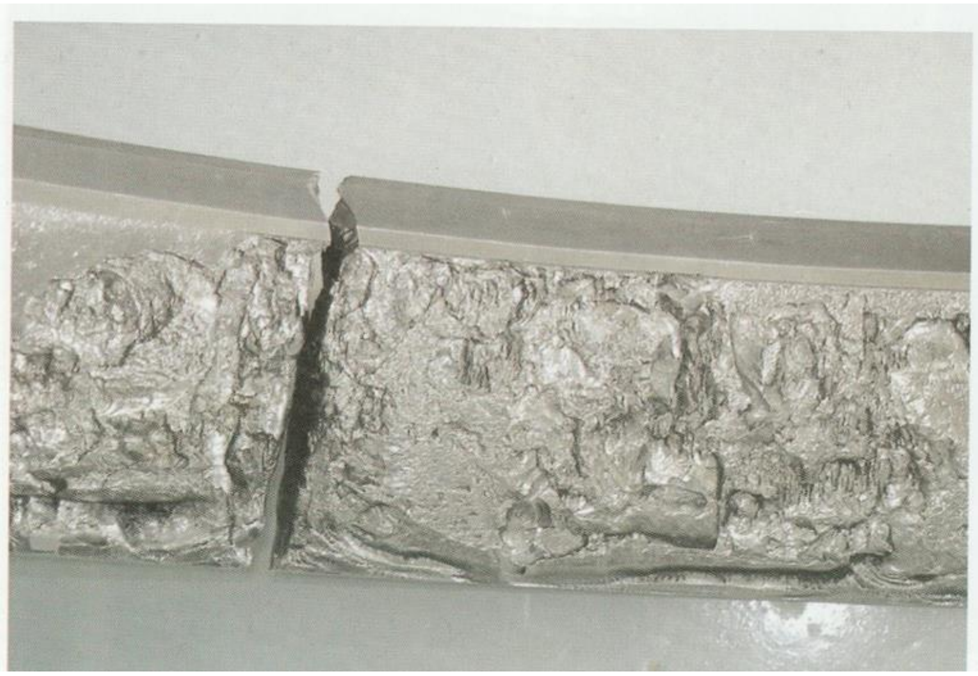


Typical wear on a lateral rolling body surface

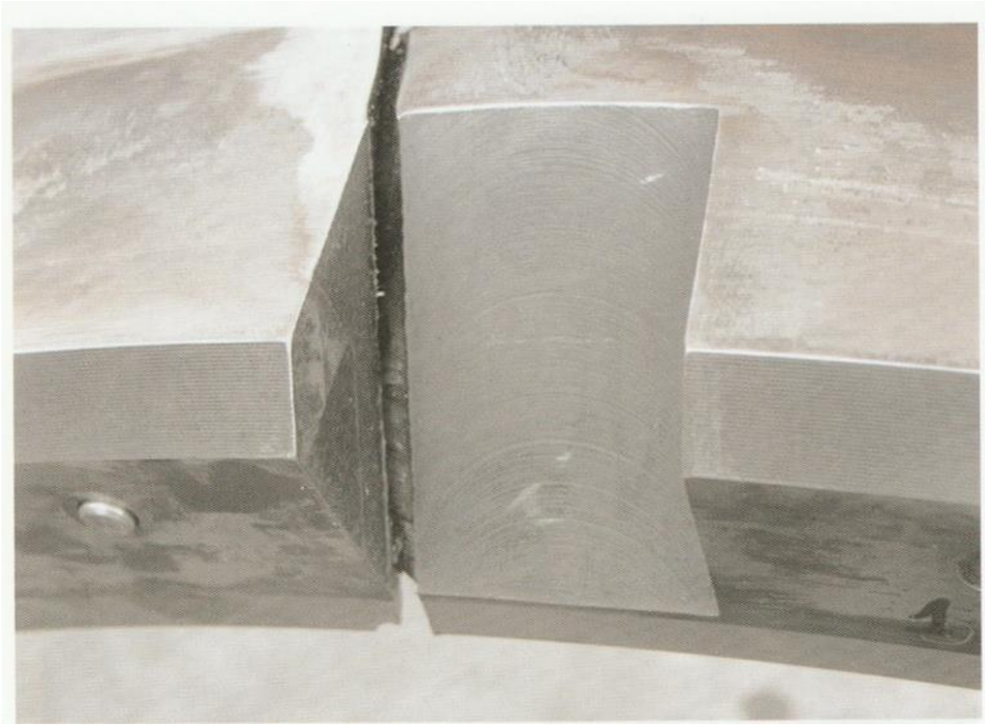
Spall of rolling bodies due to poor lubrication

Local cage and rolling bodies damage due to Spalling.

Cage wear due to the existence of water in the lubricant due to poor sealing.



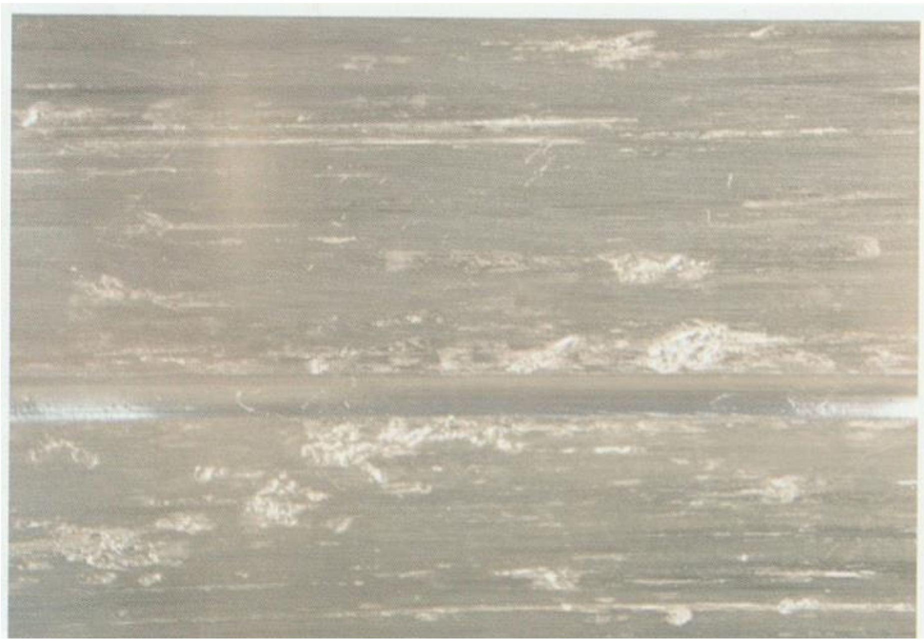
Heavy Spalling was the beginning of the ring breaking.



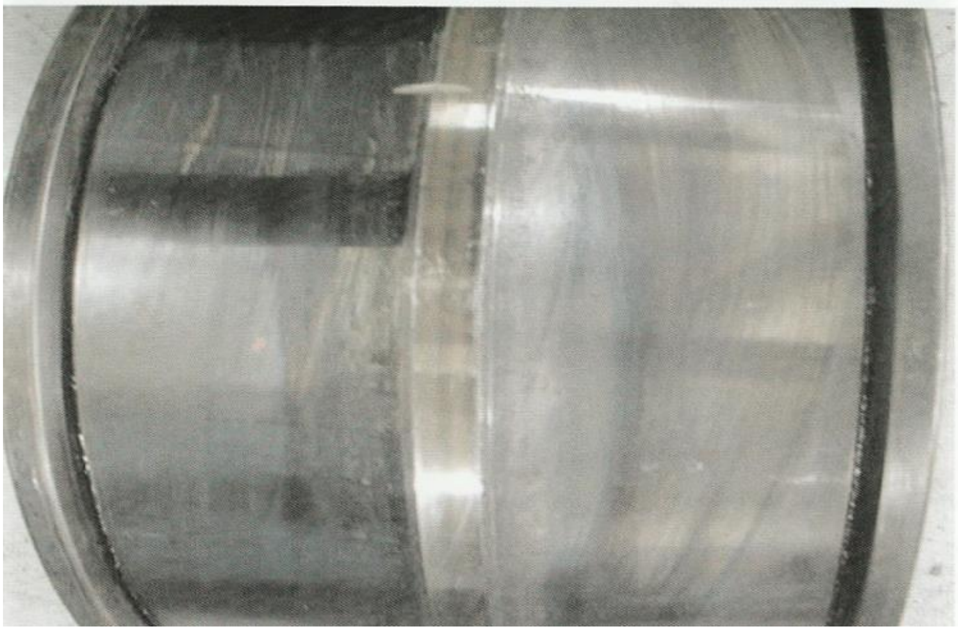
Fracture by stress gathering. High loads and poor wedge selection.

Pits due to foreign particles or by particles from Spalling of the bearing.

Ring damage due to foreign particles of hard material.



Engravings from foreign particles and small cavities.



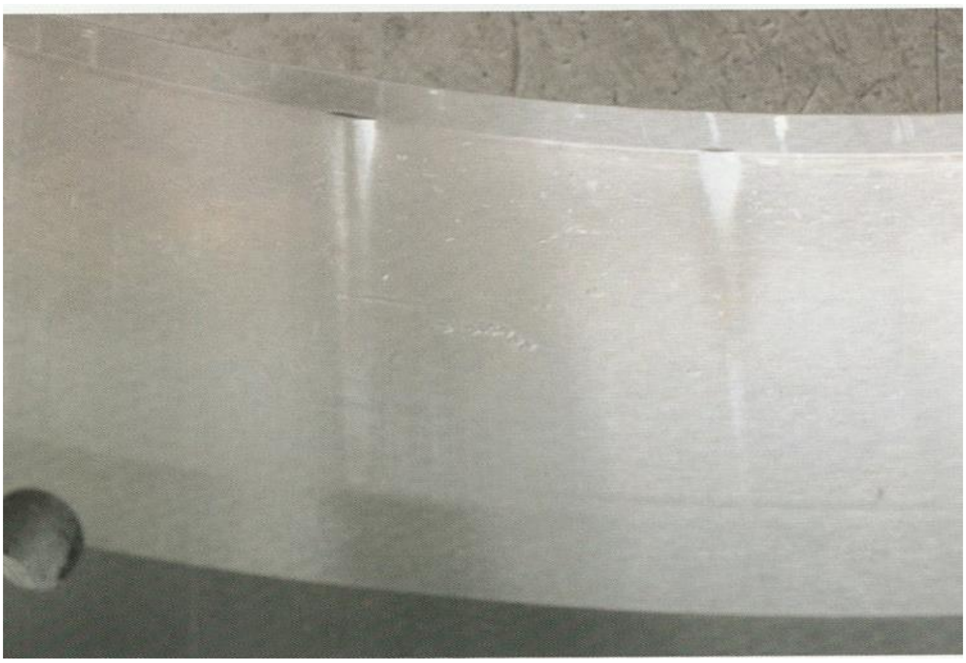
Color change as a result of overheating.

Cage disaster due to obliquities of the rolling body position in motion inverse.

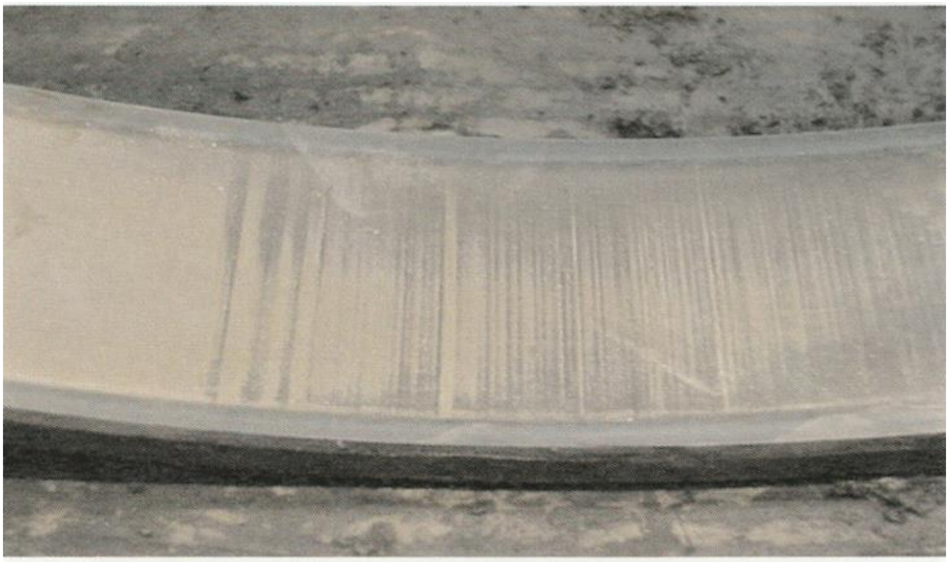
Scrolling phenomenon in lateral surface of the rolling body.

Wear from load unevenness from cone-ring misalignment.

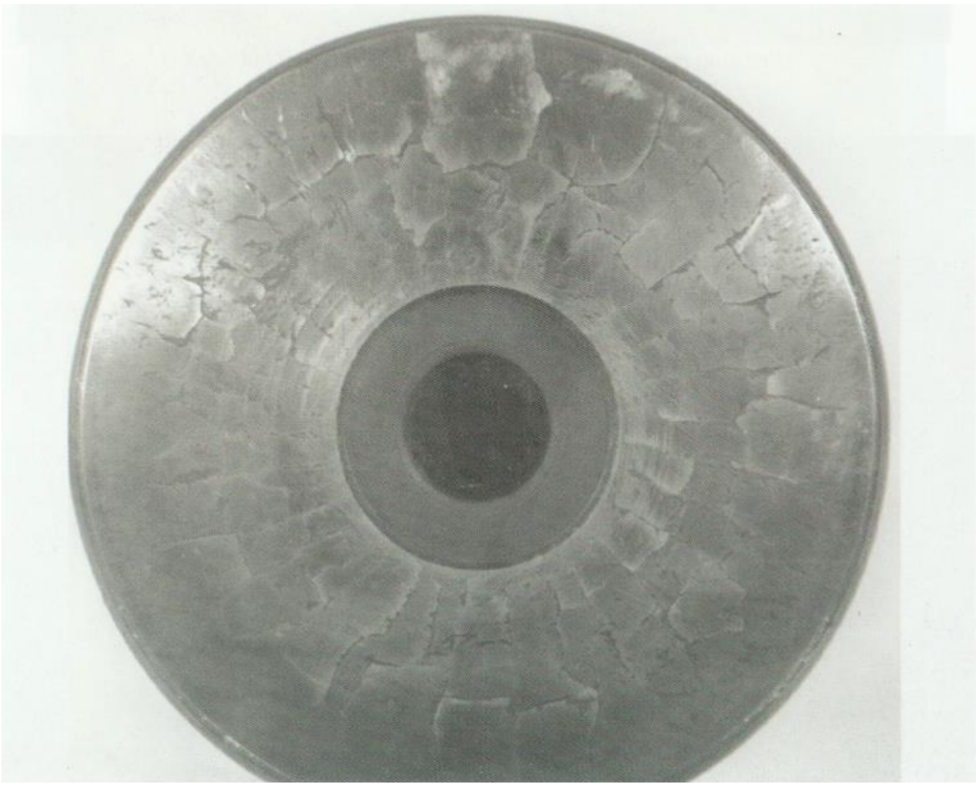
Brinelling in a stand-by machine due to external vibrations.



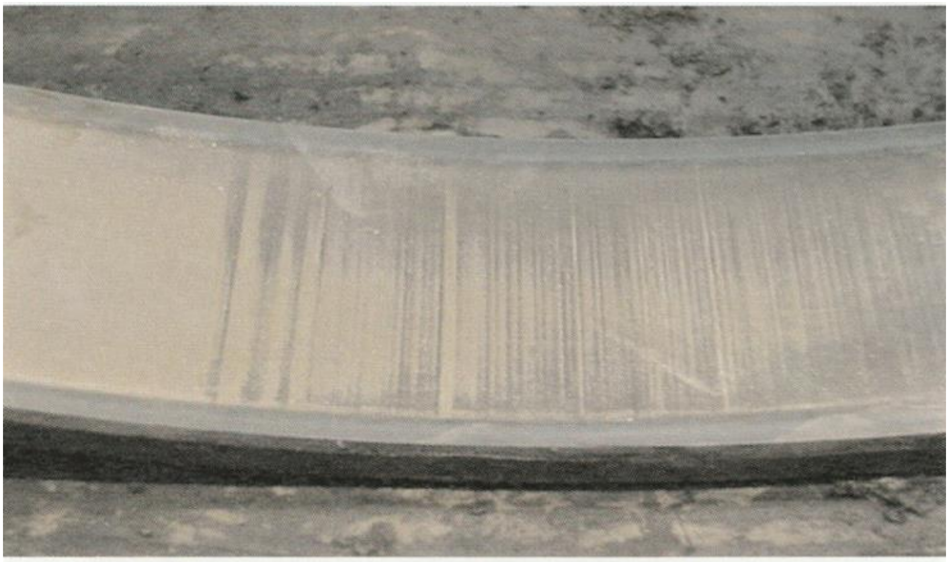
Brinelling in an operating machine due to high impact load.



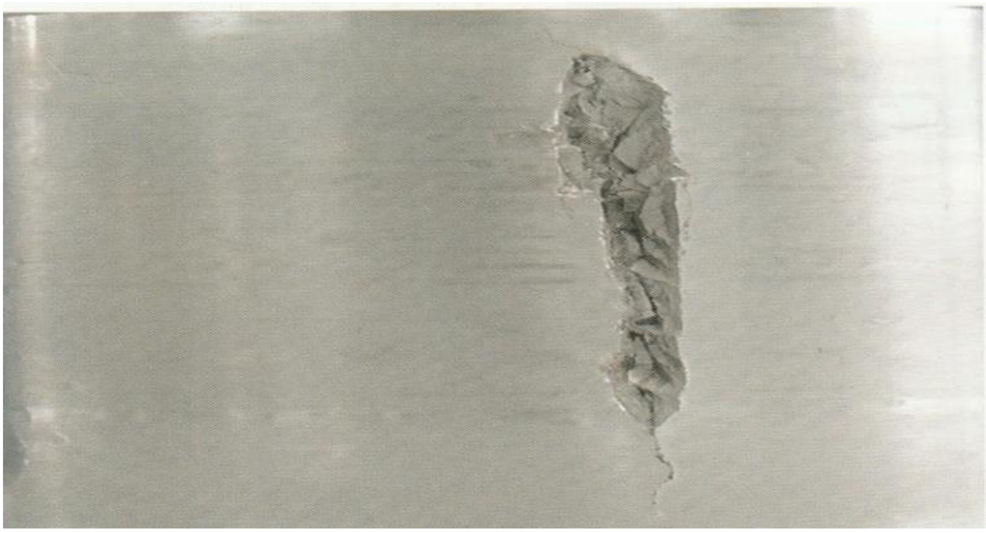
Brinelling from lubricant layer breakage and metal contact.



Roller damage by overheating from high load and marginal lubrication.



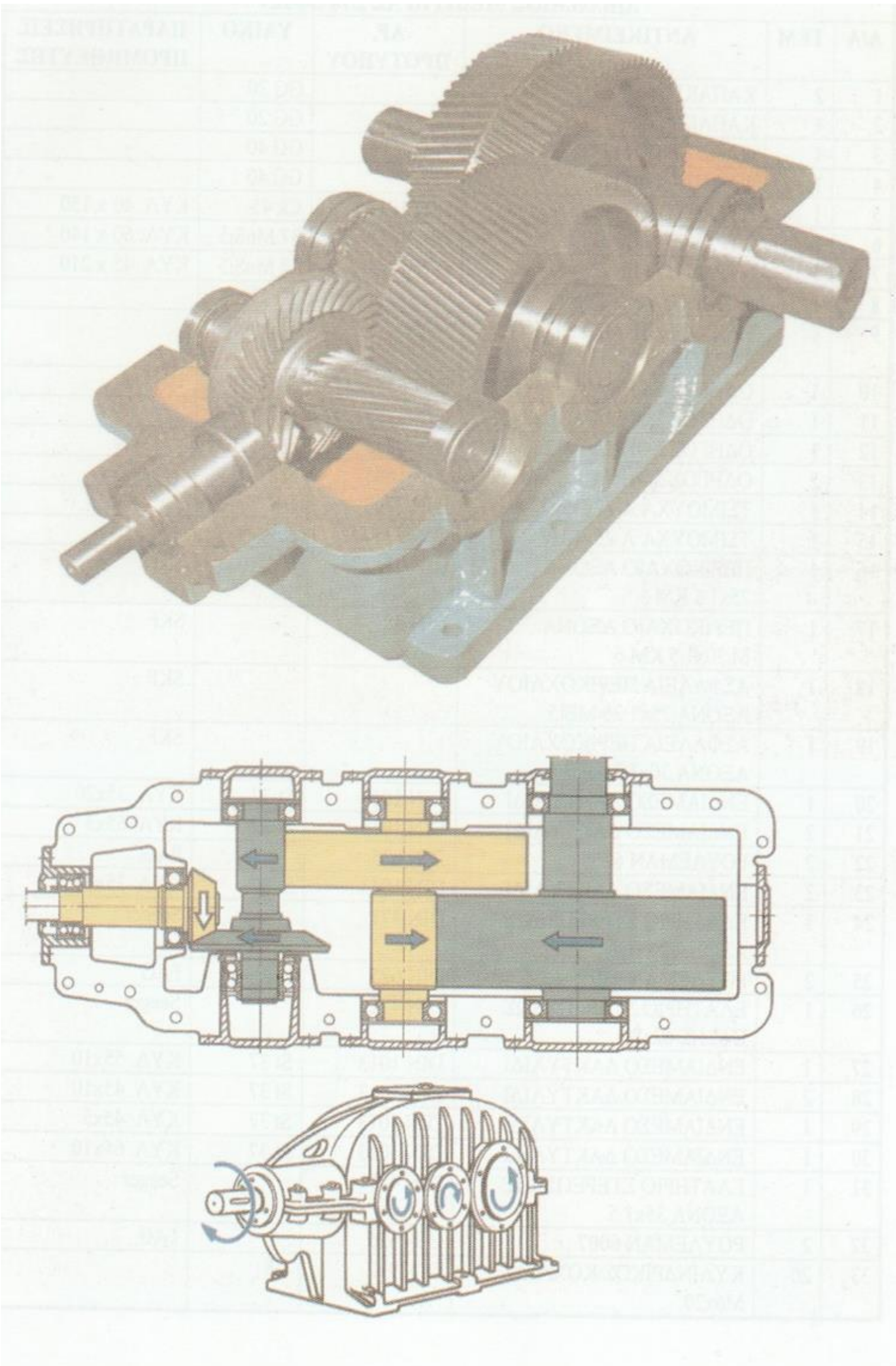
High Spalling due to high axial load.



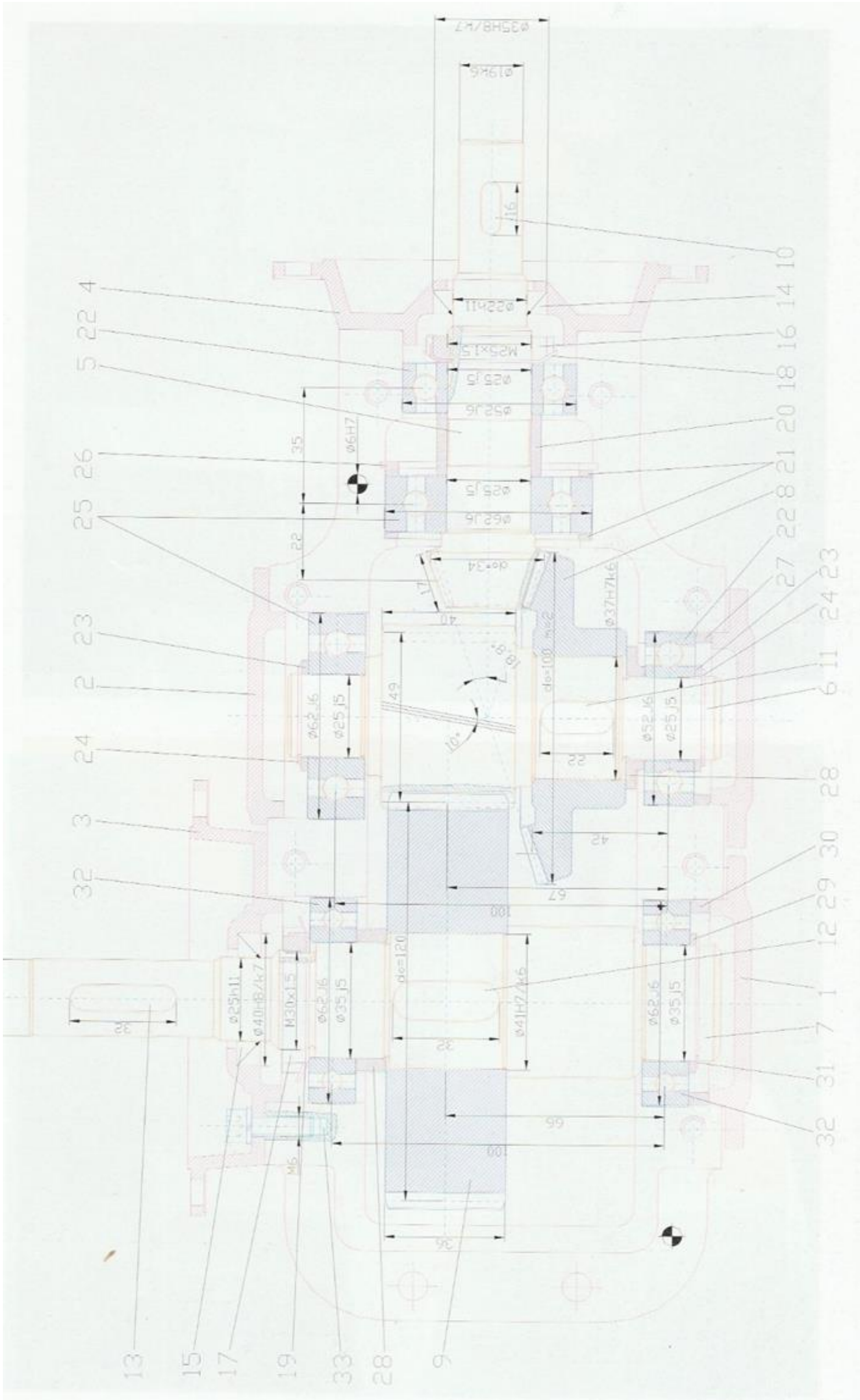
Ring disaster from the occurrence of foreign particles.

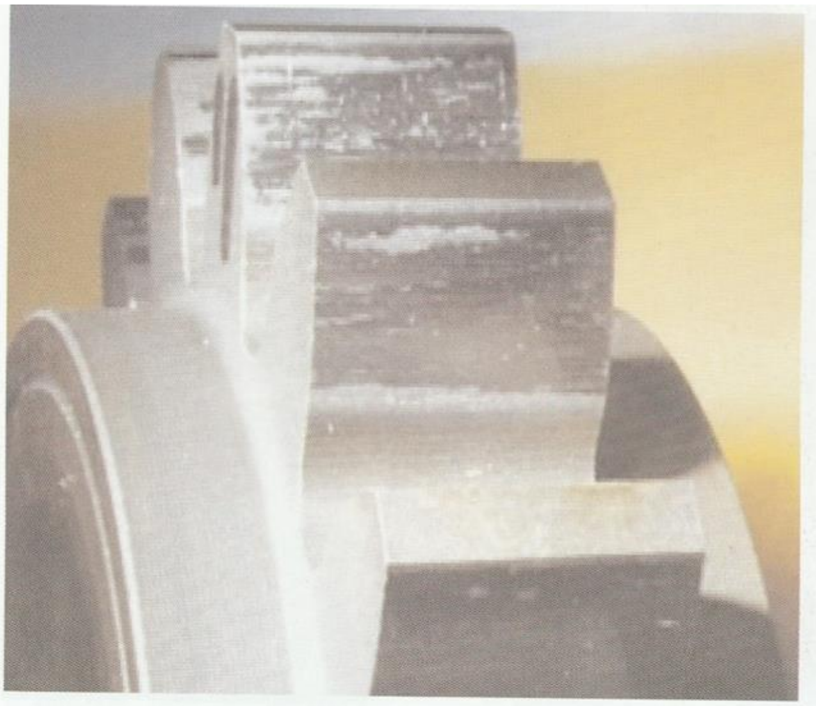
Rolling body and cage damage from Spalling.

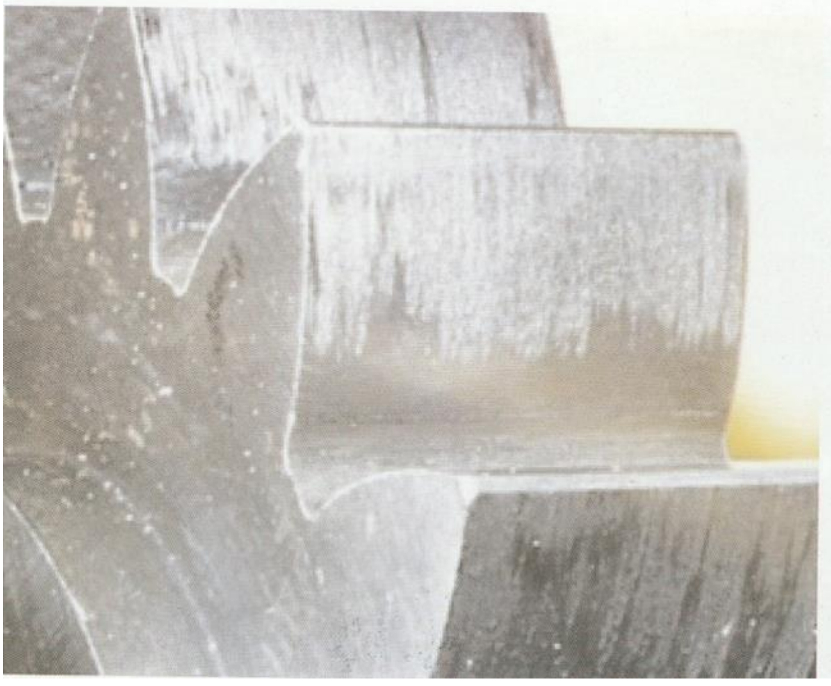
Ring breakage and uneven load due to poor bearing support.

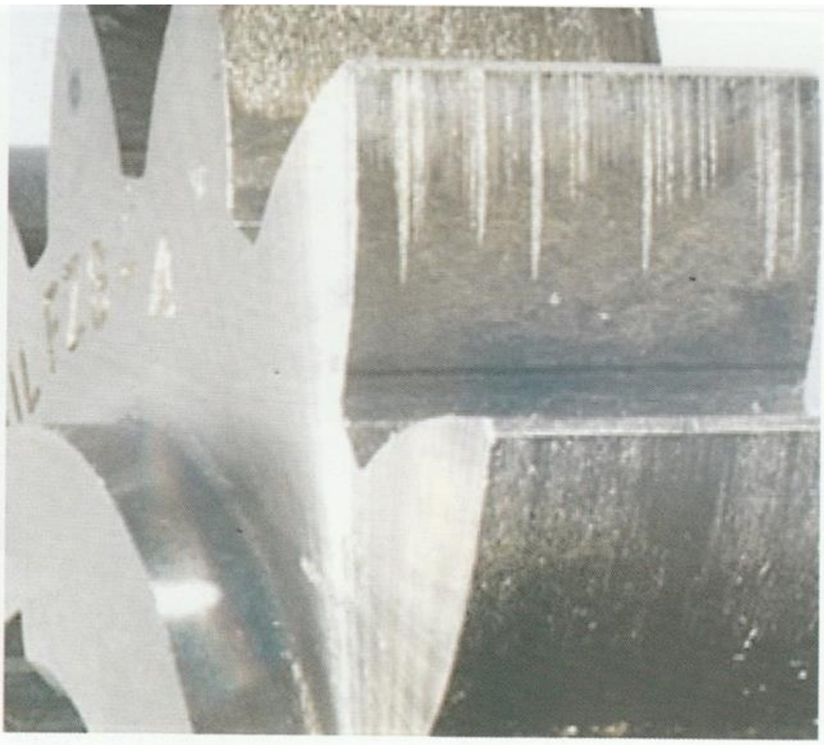


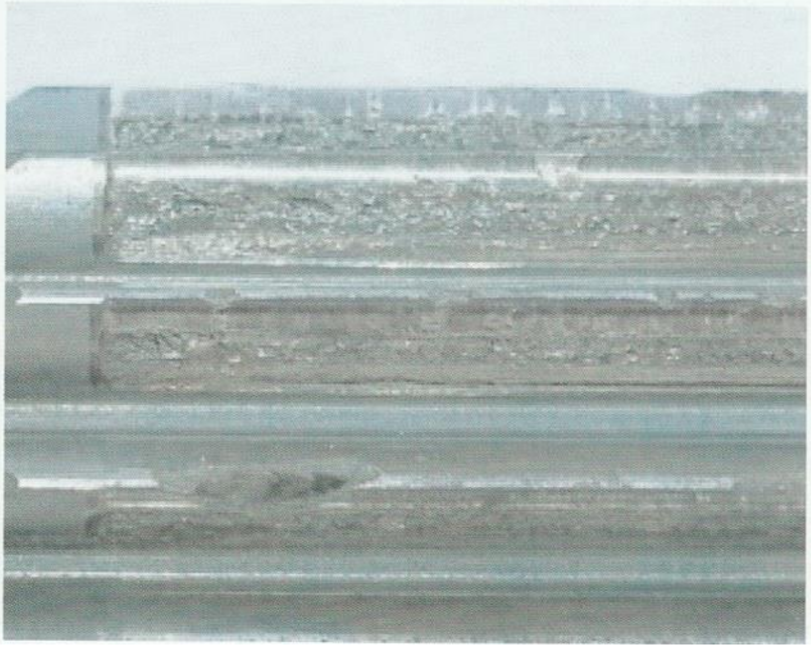
Two-speed gearbox					
S/N	PIECE	ITEM	MODEL NO	MATERIAL	NOTICES SUPPLIER
1	2	COVER		GG 20	
2	1	COVER		GG 20	
3	1	FLANGE Φ117		GG 40	
4	1	FLANGE Φ111		GG 40	
5	1	SHAFT I	DIN 1013	Ck 45	CYL. 40×150
6	1	SHAFT II	DIN 1013	37MnSi5	CYL. 60×140
7	1	SHAFT III	DIN 1013	37MnSi5	CYL. 45×210
8	1	CONICAL WHEEL		Ck 45	
9	1	HELICAL FRONTAL WHEEL 10^{o}		37MnSi5	
10	1	CHOCK GUIDE A 6×6×16	DIN 6885	St 50-1k	
11	1	CHOCK GUIDE A 10×8×22	DIN 6885	St 50-1k	
12	1	CHOCK GUIDE A 12×8×32	DIN 6885	St 50-1k	
13	2	CHOCK GUIDE A 12×8×32	DIN 6885	St 50-1k	
14	1	SEAL A 22×35×7	DIN 3760	NB	
15	1	SEAL A 25×40×7	DIN 3760	NB	
16	1	SHAFT NUT M 25×1.5 KM 5	DIN 981		SKF
17	1	SHAFT NUT M 30×1.5 KM 6	DIN 981		SKF
18	1	SHAFT LOCK NUT 25×1.25 MB 5			SKF
19	1	SHAFT LOCK NUT 30×1.25 MB 6			SKF
20	1	INTERMEDIATE RING	DIN 1013	St 37	CYL. 35×20
21	2	INTERMEDIATE RING	DIN 1013	St 37	CYL. 65×3
22	2	BEARING 6205	DIN 625		FAG
23	2	INTERMEDIATE RING	DIN 1013	St 37	CYL. 35×3
24	1	BASE MOUNTING SPRING 25×1.5	DIN 471		Seeger
25	2	BEARING 6305	DIN 625		FAG
26	1	BASE MOUNTING SPRING 62×2	DIN 472		Seeger
27	1	INTERMEDIATE RING	DIN 1013	St 37	CYL. 55×10
28	2	INTERMEDIATE RING	DIN 1013	St 37	CYL. 45×10
29	1	INTERMEDIATE RING	DIN 1013	St 37	CYL. 45×5
30	2	INTERMEDIATE RING	DIN 1013	St 37	CYL. 65×10
31	20	BASE MOUNTING SPRING 35×1.5	DIN 471		Seeger
32		BEARING 6007	DIN 625		FAG
33		CYLINDRICAL BOLT M6×20	DIN 912	8.8	

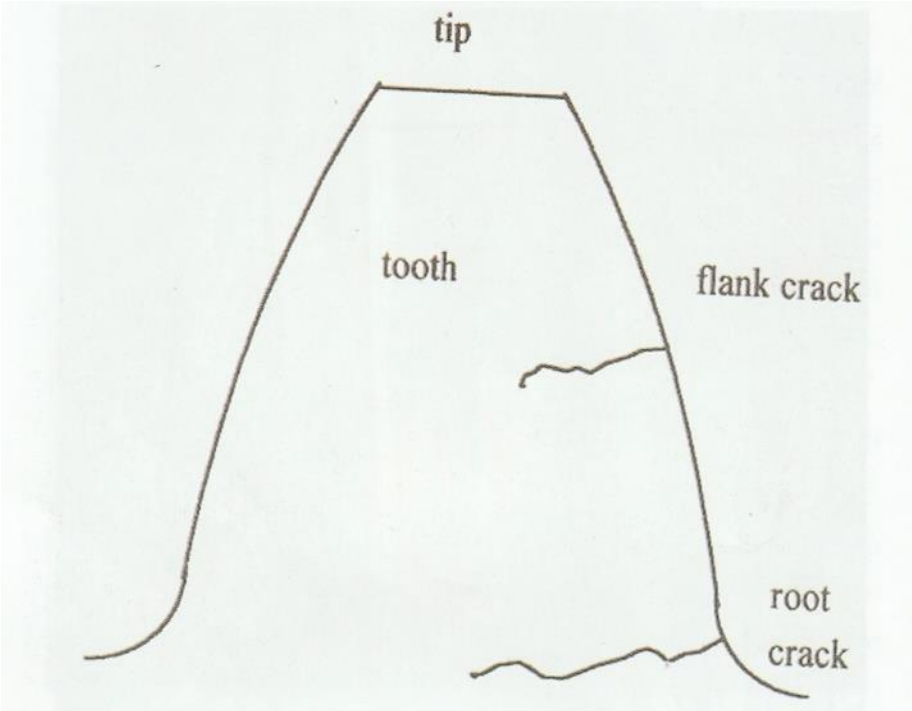








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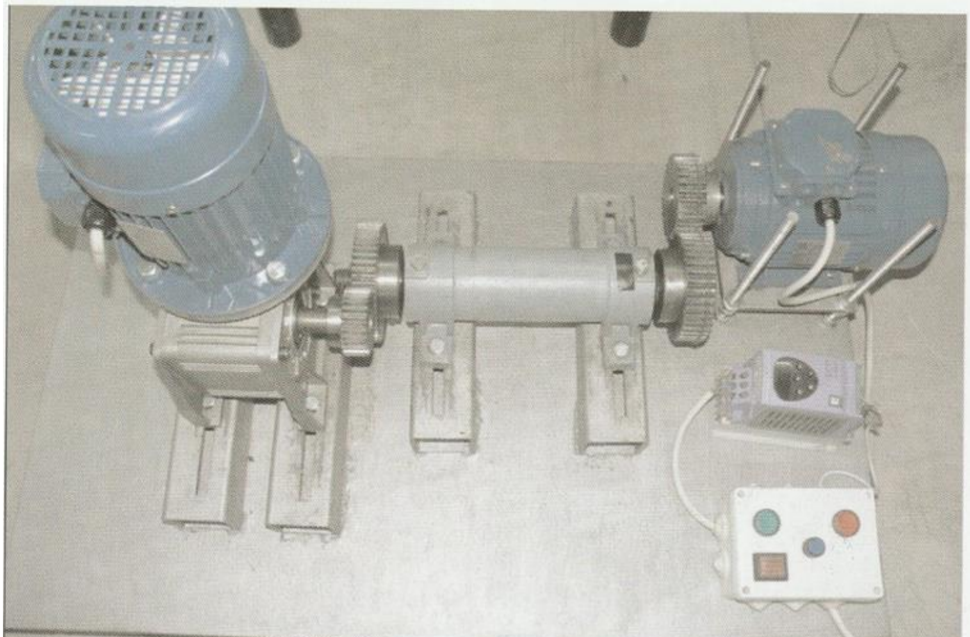
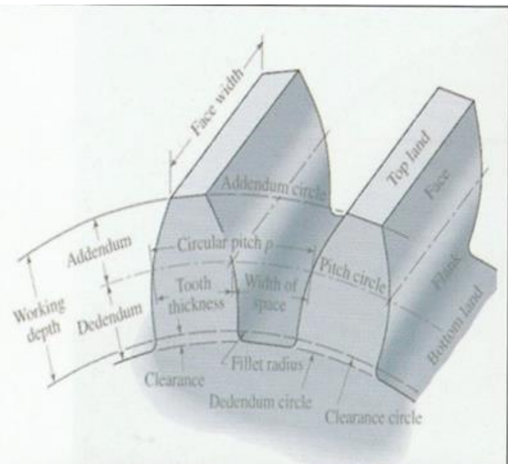
tooth
flank crack
root
crack




Face width
Top land
Addendum circle
Face
Addendum
Circular pitch p
Pitch circle
Flank
Working depth
Dedendum
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Width of space
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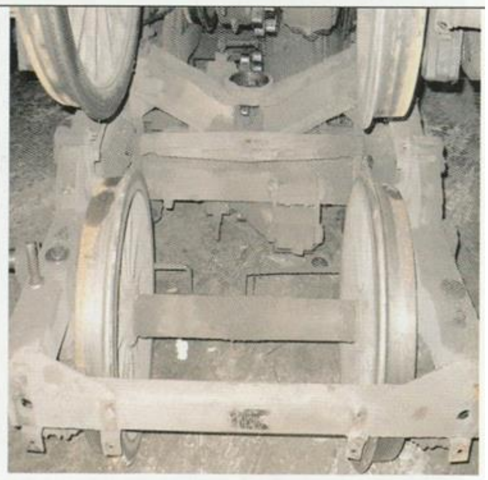
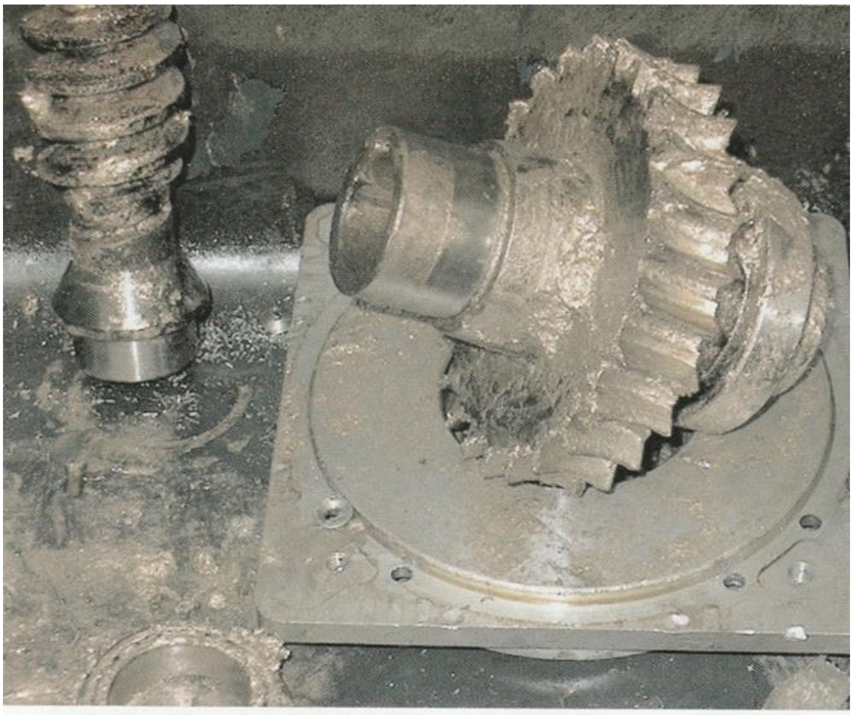
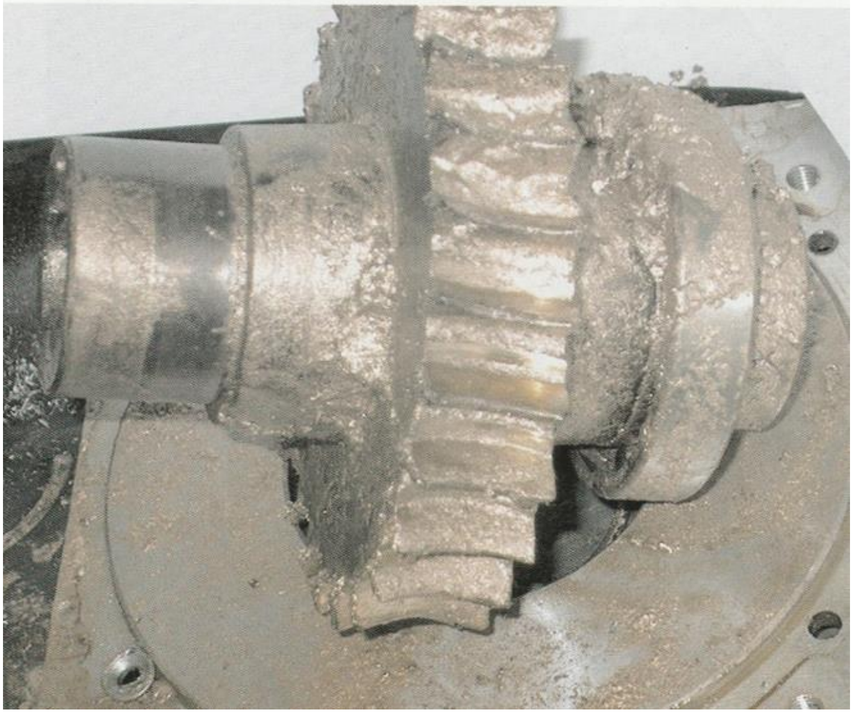
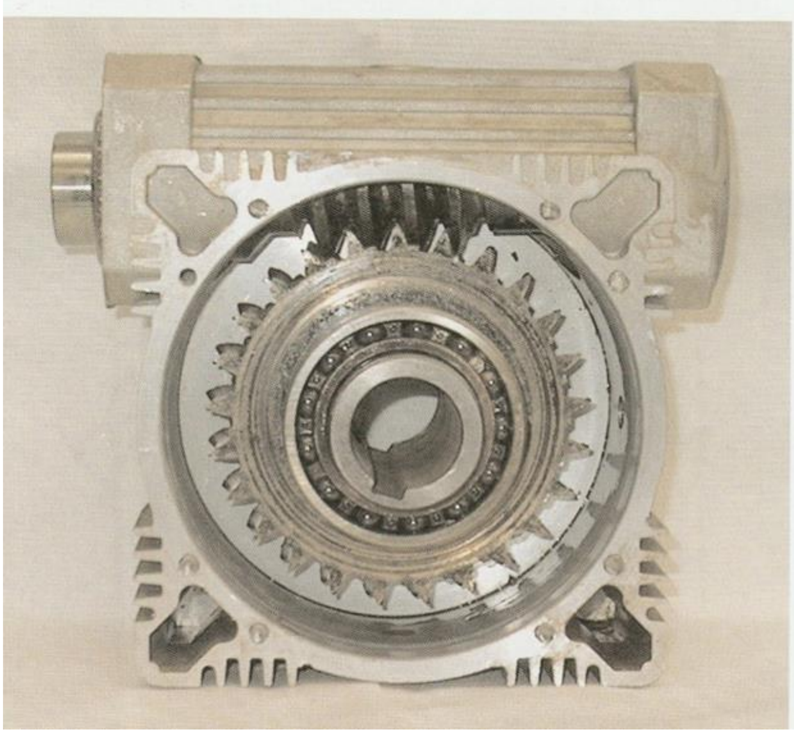
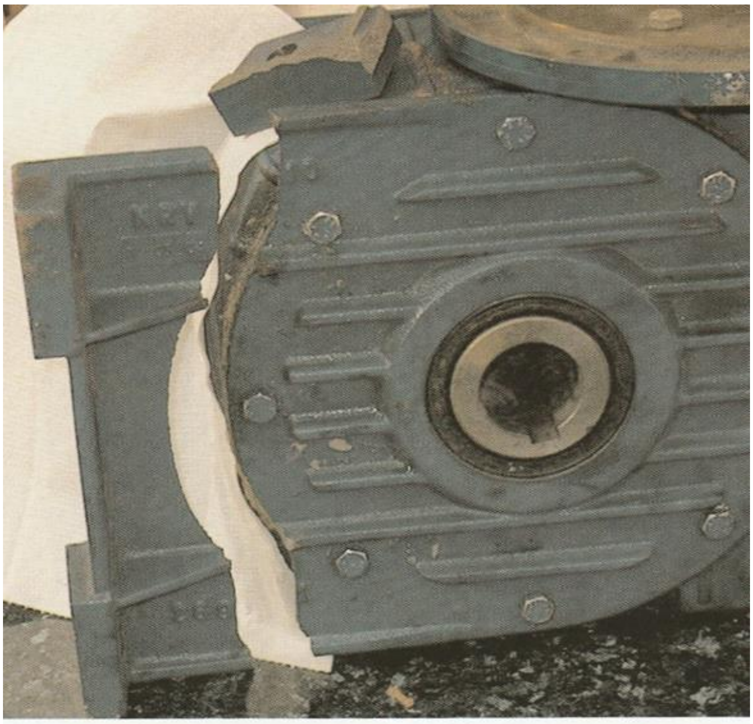
Fillet radius
Dedendum circle
Clearance circle

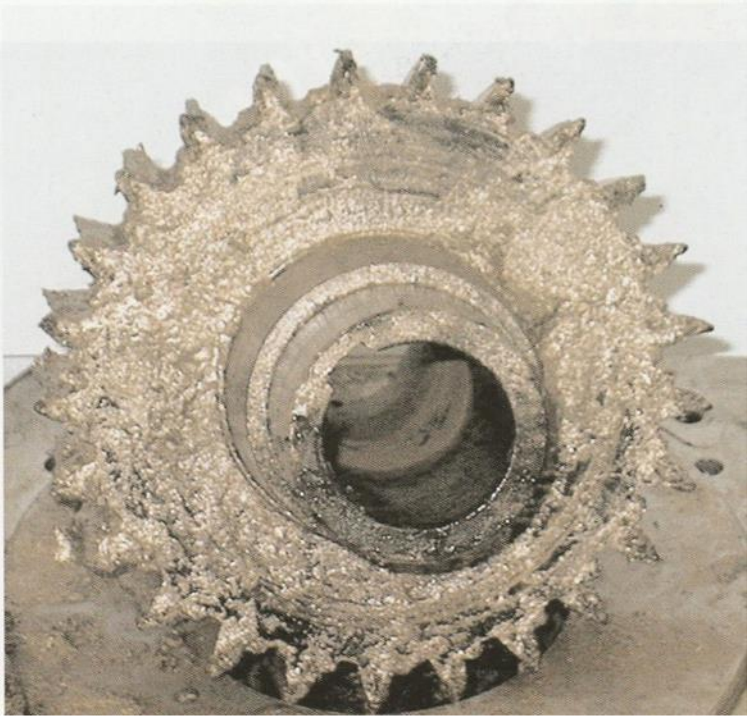


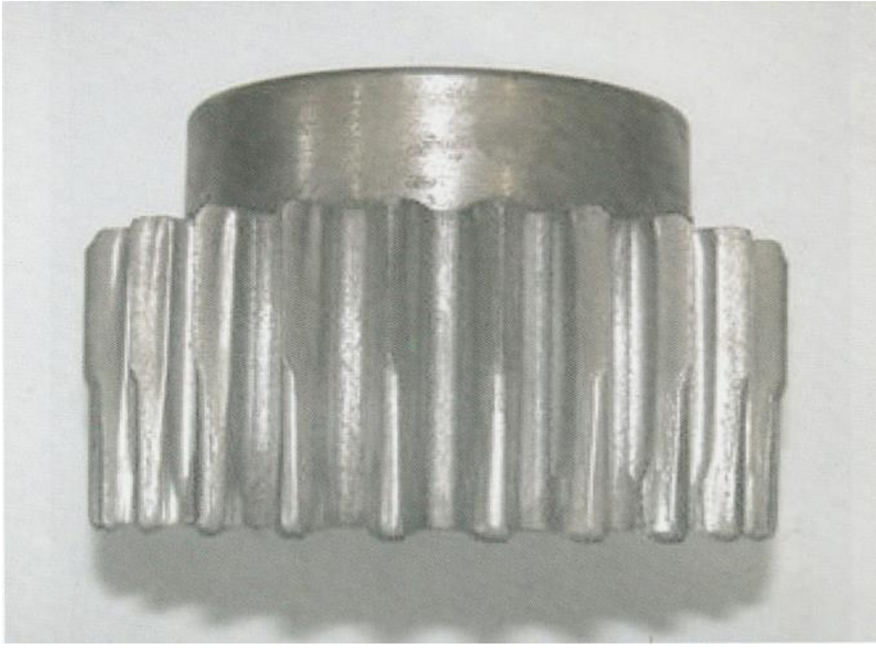




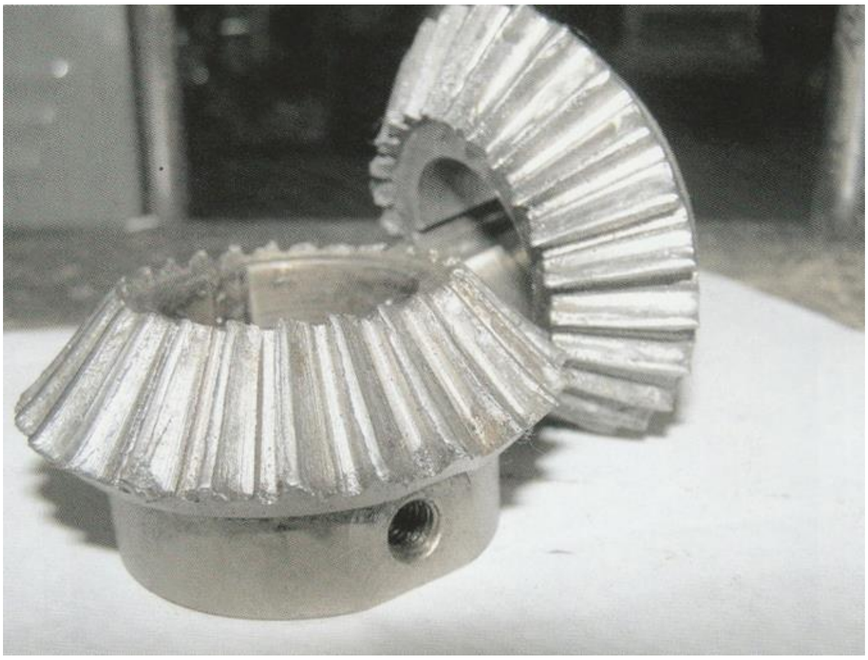




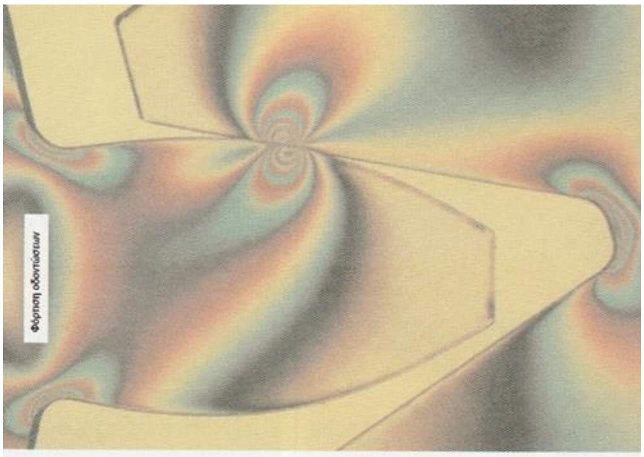




Wear on teeth of St32 cooperating wheel. Pinion of painted St50. impact loads appearance and delay in the starting of gantry.



Conical wheels of St32. cause of wear the shaft's eccentricity.


Φόρτιση οδοντώσεων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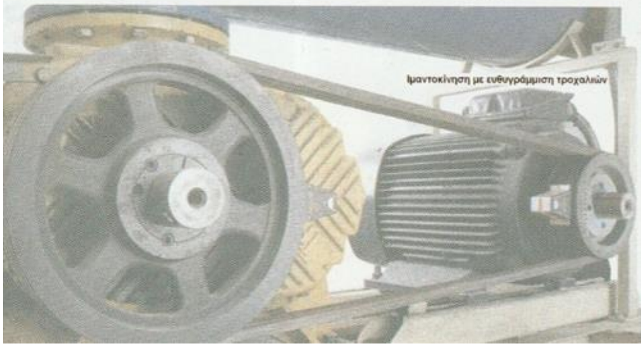
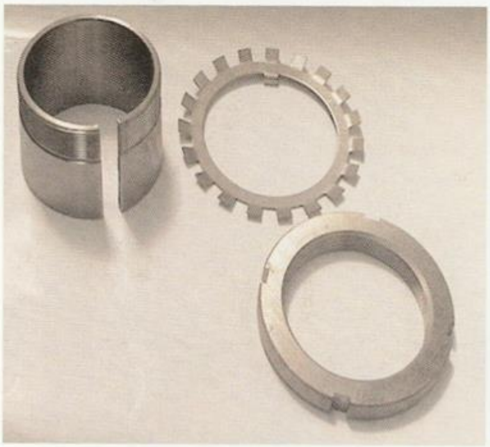
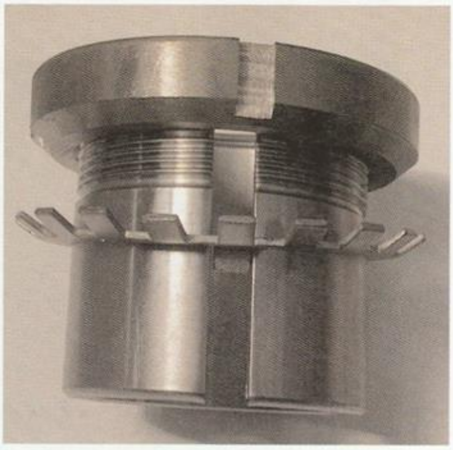
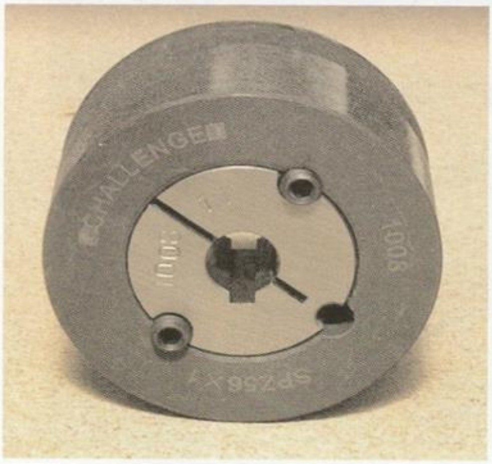

Ιμαντοκίνηση με ευθυγράμμιση τροχαλιών






CHALLENGE
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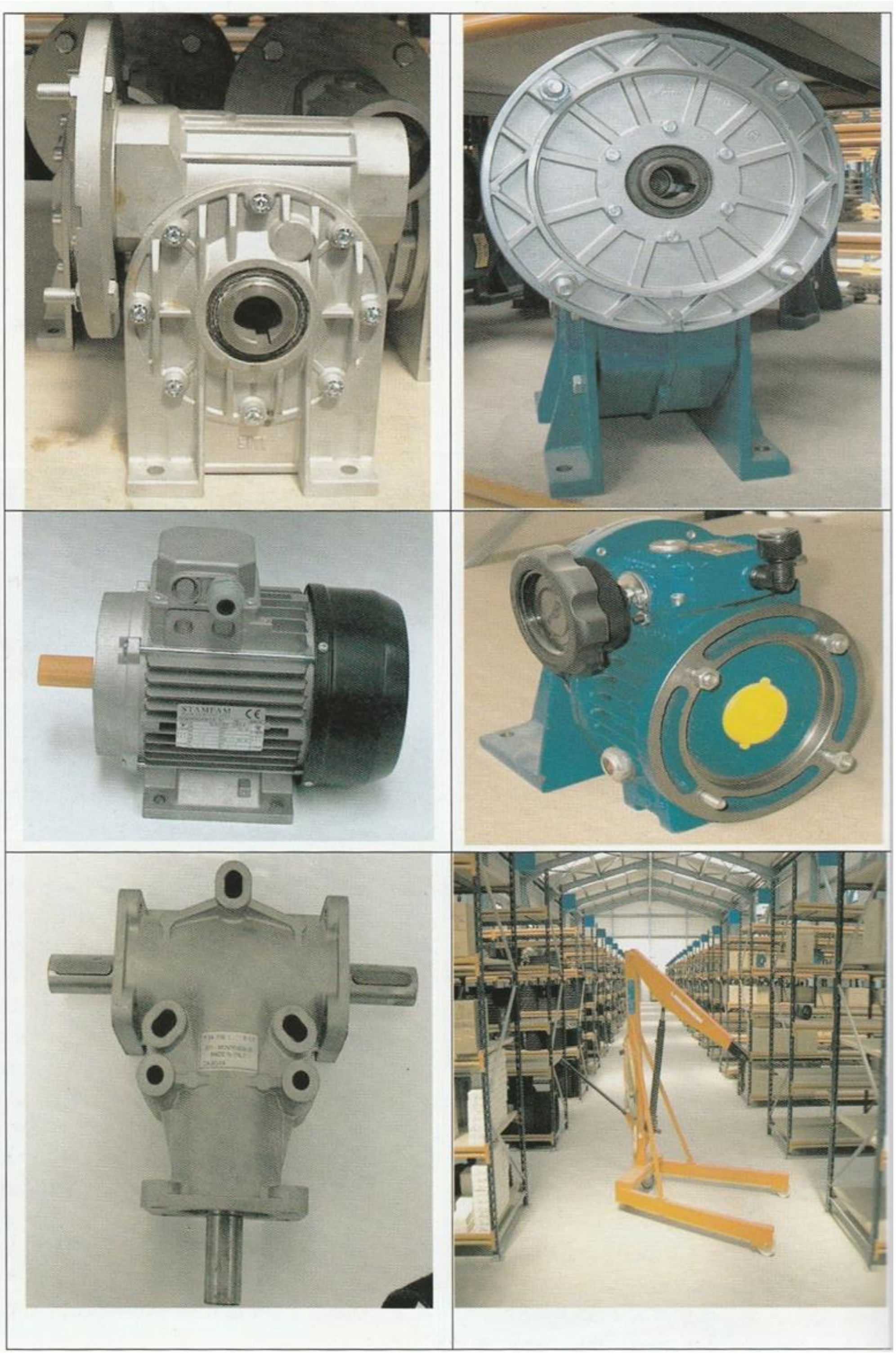
CHALLENGE
HC 280



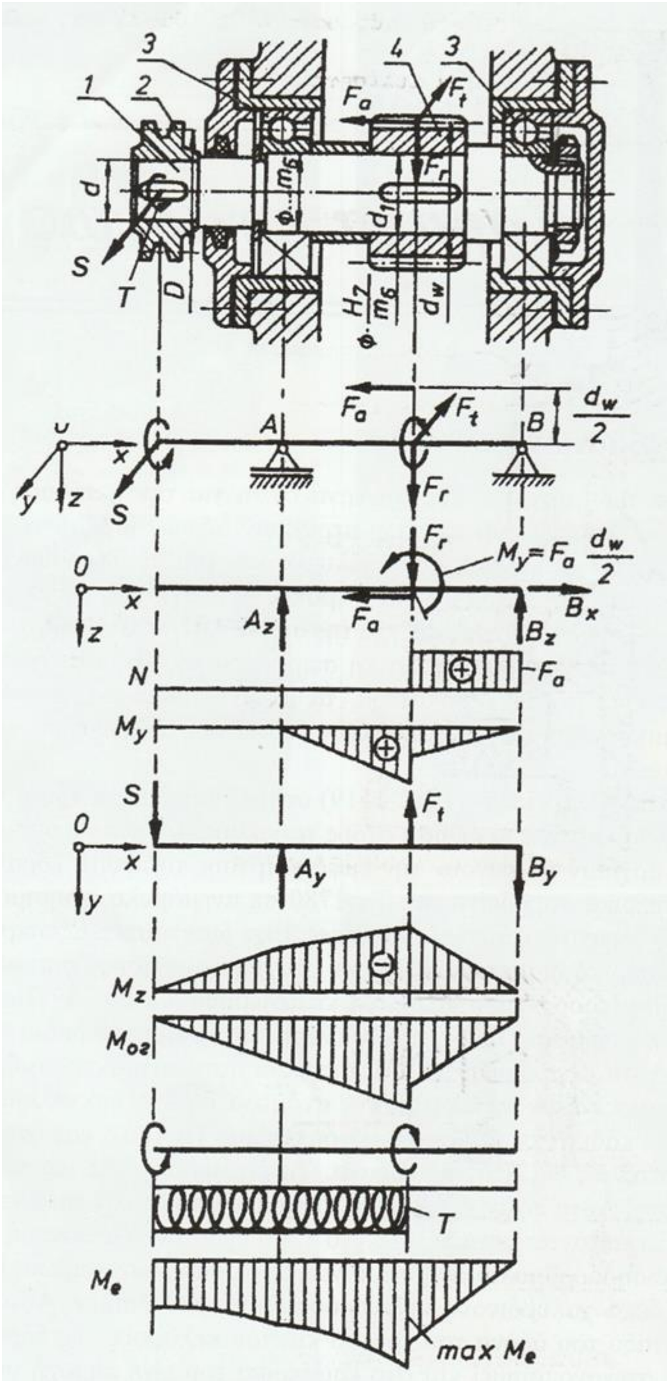
CHAPTER 3 ROLLING BEARINGS

3.1 TYPES OF ROLLING BEARINGS

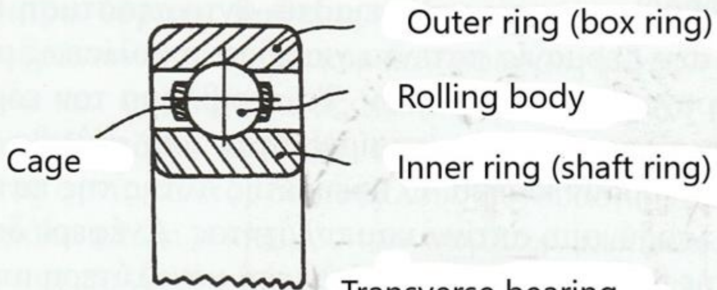
Engine components which work together for transmission and power transmission are supported on the shafts by means of links. The shafts rest on the hull through the bearings and transfer the loads to it, as in figure 3.1 where the method of calculating the shaft load, in two levels, is shown, for its control in complex stress.

The use of bearings is catalytic for trouble-free transmission and power transmission. In particular, roller bearings, commonly bearings, because of their small loss ratio compared to other bearings, were called anti-friction.

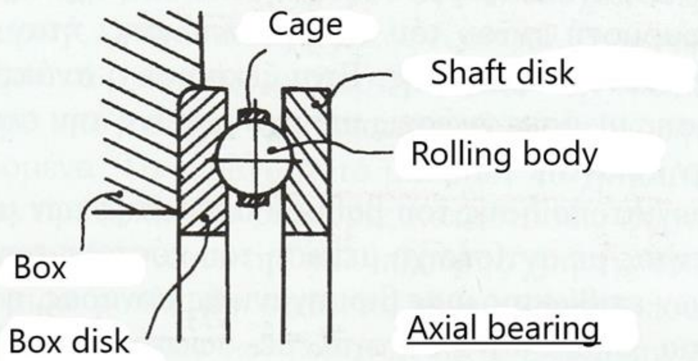
Leonardo Da Vinci (1452-1519) dealt with friction and later his manuscripts gave impetus to technicians to make better bearings. Thus began the appearance of the first roller bearings. Some examples are mentioned: in 1780 the first axial bearing with balls was used in a windmill. It had dimensions: outer diameter = 860mm, inner diameter = 610mm and used forty cast iron ball with a diameter of 57mm. In 1794 a patent for bearing was registered in England. It was described as a bearing with balls that was capable of lifting small and large loads. The rings were made of cast iron. The balls were placed between two rings, through a bore perpendicular to the outer ring. In 1802 two patents for axial bearings were registered in France. One with spheres and the other with truncated cones. These had an iron cage, which did not allow the rolling parts to touch each other. In 1860 a construction was done on the bearing of a railway vehicle wheel to reduce friction. The construction included ten cast cylinders in the diameter of 38mm. They were fitted between the axle and the bearing housing. The whole system was sealed and inside had a supply of liquid lubricant from piping at the top and then after lubricating the cylinders, they were collected at the bottom.



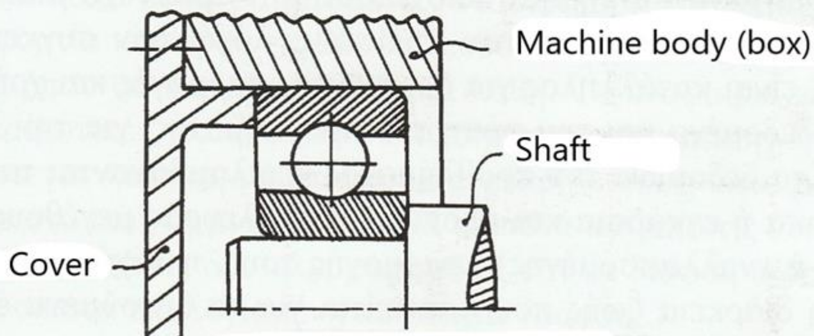
Sketch 3.1 Shaft and bearing loading.



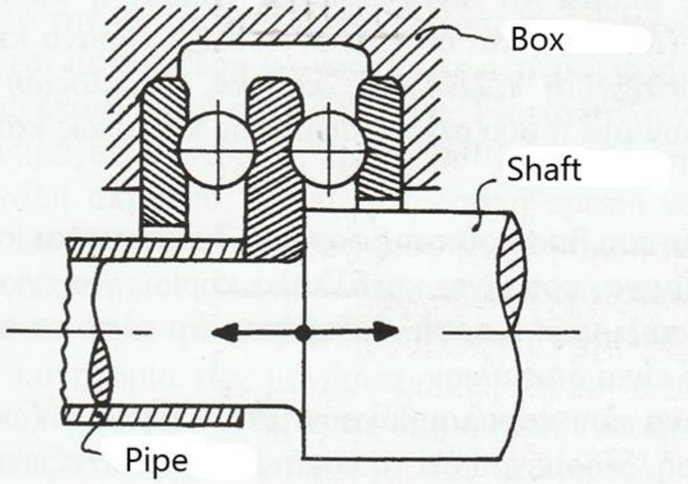
Transverse bearing



Axial bearing



Annular bearing support



Discoid bearing support

Sketch 3.2 Types and supports of rolling bearings.

These bearings traveled 20000 km without needing replacement or repair. In 1887 a patent for a rolling bearing with two rows of spheres was registered in Germany for use on railway wheels. The bearing housing was made of cast iron, while the spheres were made of BES-SEMER steel and separated by an iron cage. The inventor of this patent spoke for the first time about equivalent radius of curvature. That is, he stated that the radius of the rolling groove should be slightly larger than the radius of the sphere. Thus the coefficient of friction on this bearing was reduced. This type of bearing has been applied to many machines. The increasing demand for the application of this roller bearing was an indication of the growing need to reduce friction. Thus began the development of roller bearing technology with all the knowledge that existed at that time, for materials, design and lubrication. Today, the need to maximize the efficiency of the machines and increase productivity with a corresponding reduction in product costs is a key factor in the survival of an industrial unit, the quality of the bearings is of paramount importance. Quality with the right choice, good application and proper maintenance are the quadruple of a successful installation. Each type of rolling bearing, due to its specific characteristics, is suitable for certain applications and uses determined by the data and the nature of the problem, for the solution of which it is intended. The data of the problem include the applied loads (radial or transverse and axial) in terms of their size, manner and time (fixed or alternating) application, the speed of rotation of the rotors, the life required for the required bearings, the available radius or axial space etc. Roller bearings consist of four elements: the roller bodies, the roller body cage and the inner and outer element. The rolling bodies may be balls or cylinders, cones, crankshafts, etc. and so the corresponding rolling bearings are called ball bearings or cylindrical, conical, barrel-shaped etc. They may have one or two rows of rolling bodies and thus be simple or double (two balls).

Depending on the direction of their base load, they are distinguished, sketch 3.2 in:

Transverse or radial bearings, that are mostly suitable for transverse (radial) loads and have an annular shape, since both the inner and outer elements are rings.

Axial or thrust bearings which receive only axial (thrust) loads and have a disc-shaped form, since the inner and outer element are discs.

Combined loads that receive both transverse and axial loads and are either annular or disc-shaped.

The most basic types of roller bearings are given below.

Deep groove ball bearings, sketch 3.3, are now a common type of bearings and are widely used when there are only transverse or transverse loads and no large axial loads. Because the loading capacity of a bearing depends on the number of rolling bodies, deep groove bearings come in a variety of combinations and designs. In this important role plays the cage of rolling bodies, which must be one-piece and very strong. There are single and double row bearings. In the double row they have a common inner and outer element in which there are two sockets where the two rows of balls work. This type of design ensures resistance in high transverse and several axial loads, regardless of the load direction, in a place smaller than the two independent single row bearings require for the same data. These bearings can be used in pairs or three together to deal with different charging situations˙ they are suitable for high speeds. Due to their versatile use and low price, ball bearings are the most widely used. The angular self-alignment of these bearings is minimal so the bearing positions must be well aligned. Bipolar bearings can accept axial loads in both directions but are not suitable for cases where there will be angular errors. Closed type ball bearings with protective plates (non-contact sealing) or with sealing plates (contact sealing), allow simple constructions. There are bearings closed on one side and closed on both sides. The closed bearings on both sides as well as the two-ball bearings must be filled with a correctly calculated amount of grease.

Angular contact ball bearing, sketch 3.4. These bearings are used in cases of one direction high axial loads. The contact angle between the balls and their working path is designed to be larger than the corresponding contact angle of the deep groove bearings, thus ensuring greater capacity in axial loading. The angular contact ball bearings are not detachable. They achieve high speeds and in order to reach maximum speeds their bearing positions must be machined with great prediction. Great care should be given in their proper lubrication too. The angular contact bearings, used in pairs, create stable bearings of high axial loads in both directions. Four point contact ball bearings seem to belong to the category of single-ball angular contact bearings and accept axial forces in both directions. In axial section, the shape of the rolling orbits of the inner and outer ring resembles circular arcs. The inner ring of the four-point ball bearing consists of two parts (detachable), which allows the bearings to be supplied with many balls.

Forces and axial forces in both directions. It is particularly suitable for bearings that require stable axial guidance. The small double-row angular contact ball bearings, which do not have rolling element filling notches, can accept large axial loads in both directions in proportion to their dimensions. The large double-row ball bearings have filler notches on one side with scroll elements. These bearings must be assembled in such a way that the largest load is accepted by the rolling tracks that do not have filling notches. For large axial forces of alternating directions there are double-ball contact ball bearings with split inner ring.

Double-ball self-adjusting roller bearings, sketch 3.5, can deal with large shaft blockages. These bearings have balls in two rows. With proper machining of the working path in the outer ring, the path acquires a spherical shape and the rolling elements can self-adjust and smooth out any small alignment errors, shaft bends and bearing deformations. However, this reduces the loading capacity of the bearing. The watertight double-ball self-adjusting bearings have sealing plates on both sides (contact sealing) and are supplied with a sufficient amount of grease during their construction. The angular deviation of the double-ball self-adjusting bearings can reach about 3º on each side, while in the sealed bearings the maximum is 1.5º.

Cylindrical roller bearings, sketch 3.6, have cylindrical rolling bodies that roll in cylindrical slots. The rolling bodies are approximately one in diameter in length and have rounded ends for relief from high stress. These bearings can only receive transverse (radial) loads and are commonly used in free bearings where thermal expansions and, due to loads, spindle displacements are received. Specially designed cylindrical bearings can receive small axial load. Cylindrical bearings, with some exceptions, are detachable, which facilitates their assembly and disassembly. Both of its bearing rings can be assembled with a tight fit. The hole (inner diameter) of the double row bearing may be cylindrical or conical. The special linear contact between cylinders and rolling tracks prevents the creation of stresses at the ends. The angular self-adjustment of the cylindrical bearings is very small to zero. In fact, the larger the bearing width, the smaller the values of the bearing self-adjustment angle. The mounting positions of the double row cylindrical bearings must be very well aligned. High shaft speeds are achieved with cylindrical bearings. For bearings of especially large loads cylindrical bearings of a large number of cylinders are used. The barrel rolling bearings have a double row of rolling bodies and in addition to the very large radial loads they can receive axial loads in both directions of loading. They can also be self-regulating due to the spherical shape of the routes. They are suitable for the most difficult stresses, but not suitable for high speeds. They have two rows of symmetrical barrels that adjust automatically on their own to the spherical track of the outer ring, thus, equalizing shaft bends and alignment errors. There are different types of barrel bearings. Barrel bearings, when they can by design allow the close adhesion to the barrels with the rolling wheels, then they achieve uniform distribution of stress and high load capacity. For very difficult bearing, e.g. vibrating machines, there are specially made barrel bearings with limited dimensions and radial clearance. Finally, double row barrel bearings may have a cylindrical or conical internal open. Barrel bearings, depending on their type and load, can correct angular deviation of up to 2.5º. They are sensitive to unpredictable axial movement of the shaft.

The tapered roller bearings, sketch 3.8, are designed in such a way that the lines of the tapered surfaces pass through a common point on the bearing axis.

Due to their conical and large contact surface they can receive high loads, both radial and axial. Usually used at low speeds. Tapered bearings, with the exception of closed tapered bearings are detachable bearings. The two parts can be assembled separately. The special linear contact between the cones and the rolling tracks does not allow the creation of stresses at the ends. Because tapered bearings only accept axial forces in one direction, they are usually combined with a second tapered bearing that receives axial forces only in the opposite direction. The bearing assembly is in pairs in X arrangement. The axial clearance of the bearing pair is adjusted by a spacing ring between the outer rings. With such a pair of bearings the speeds of the individual bearings are not usually achieved, which is why the speed limits are lower. Due to the linear contact of the rolling bodies in the color cones, the angular self-regulation is very small. The bearing positions of the tapered bearings must be well aligned. Closed-type tapered bearings with more roller bearings are commercially available in ready-made units per pair for O-layout.

The thrust bearings, sketch 3.9, have the same characteristics as the rest of the bearings, except that the working paths of the roller bearings are approximately perpendicular to the axis of the bearing and not approximately parallel to it, as is the case with the transverse bearings. The rolling bodies may be spherical, cylindrical or barrel-shaped and the bearings may be single or double.

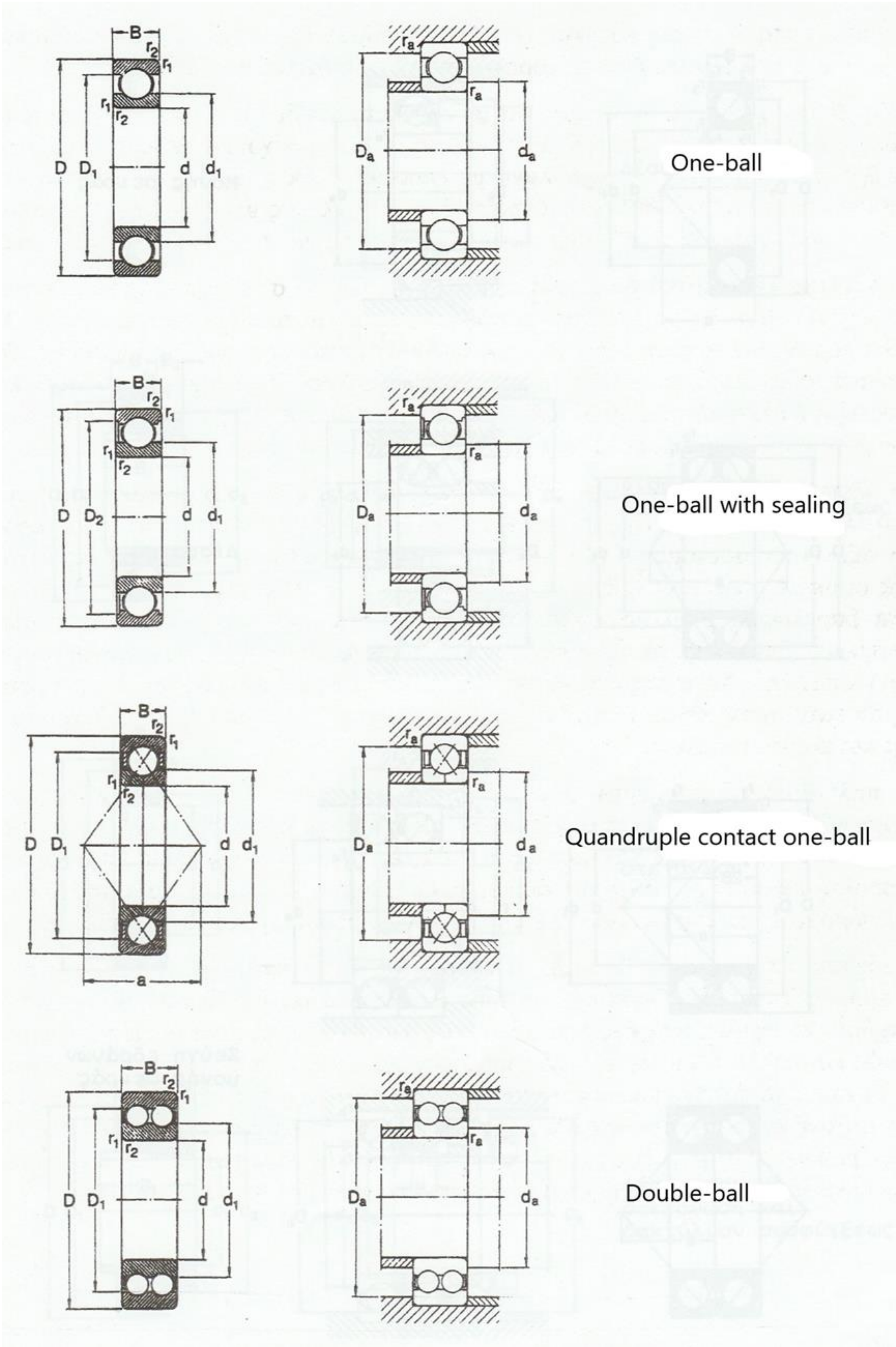
Axial ball bearings are made in two variants: single and double energy bearings. Both variants accept large axial forces. In addition to the flat plate variant, there are axial ball bearings with spherical seat plates and support plates. The support surfaces of the bearing plates must be parallel to each other.

Axial cylindrical bearings are suitable for fixed bearings, capable of accepting large loads and not being affected by impact loads. The simple shape of the bearing fittings allows for many combinations. Axial cylindrical bearings are used when there are not enough axial ball bearings or other bearings. The simple geometric shape of the bearing components allows very high construction accuracy.

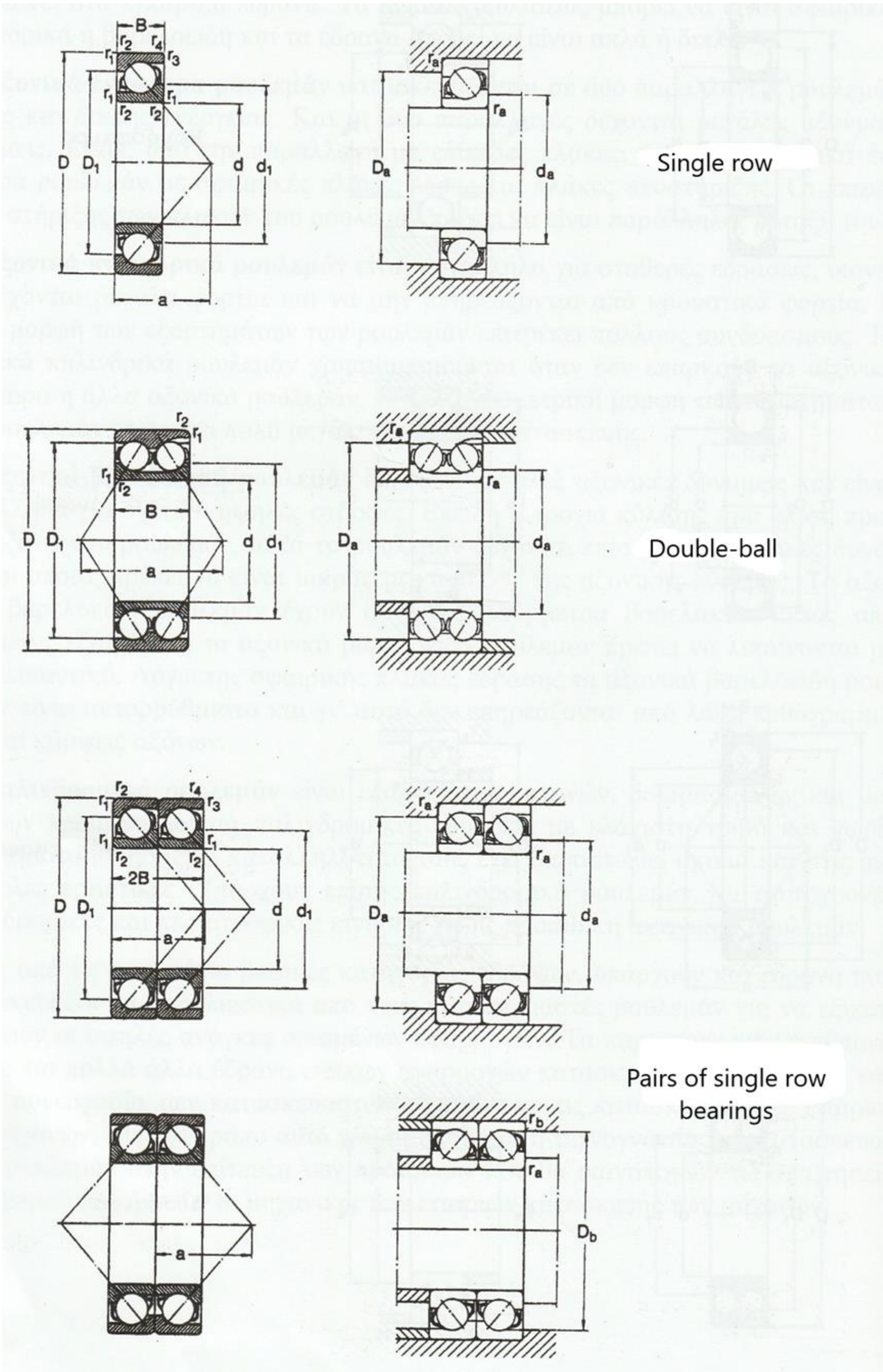
Axial barrel bearings accept large axial forces and are suitable for relatively high speeds. Because the roller bearing is inclined to the bearing axis, these bearings also receive radial forces which must be less than 55% of the axial force. Axial barrel bearings usually have asymmetric barrels. With some exceptions, axial barrel bearings are self-adjusting and therefore are not affected by alignment errors and shafts bends.

The reciprocating bearings are machine parts, tested and modern constructions for reciprocating bearings with minimal friction and without movement problems. Their suitability has been established even in the most difficult practices. There are also reciprocating bearings for simultaneous reciprocating and rotating movements without the addition of a radial of a radial bearing.

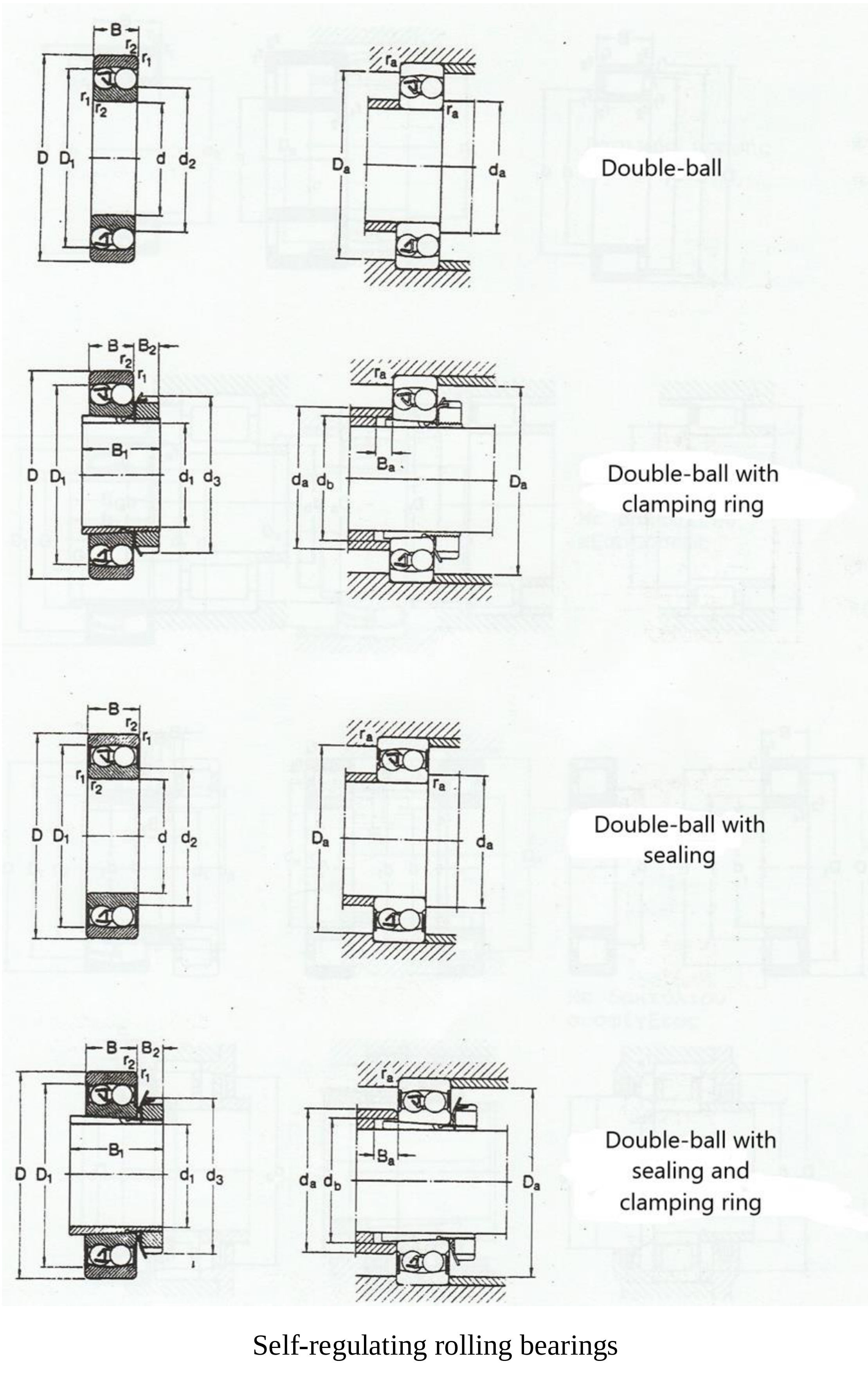
In addition to the above basic bearing categories, there are bearings manufactured exclusively by bearing manufacturers to meet the high needs of the certain applications. The above special bearings as well as many other bearings of special applications are usually manufactured in the cooperation of manufacturers and used in the achievement of the products that will meet the design requirements requested by the engineers of the machine manufacturing companies.



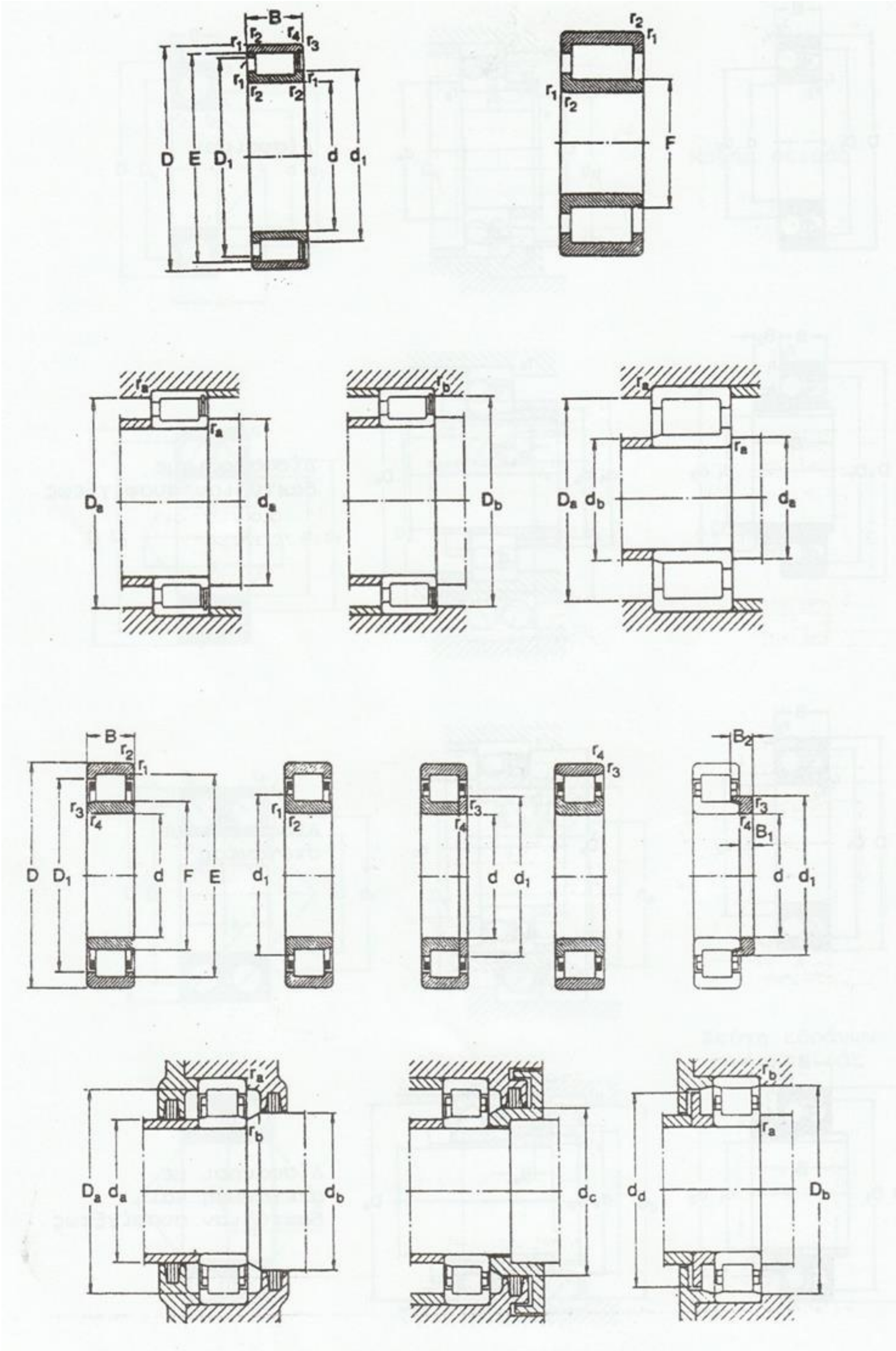
Rolling bearings of deep annulus.



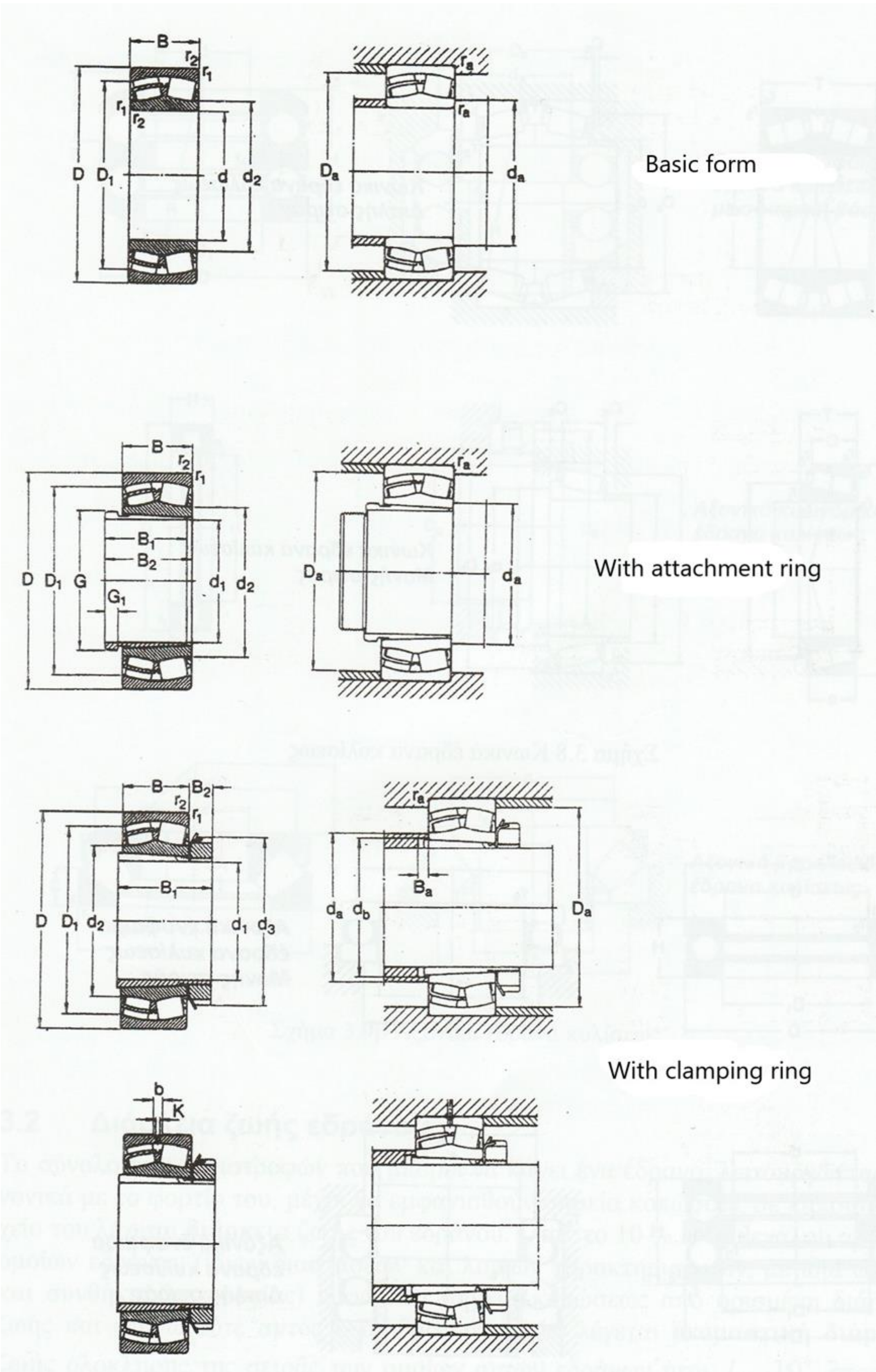
Angular contact rolling bearings



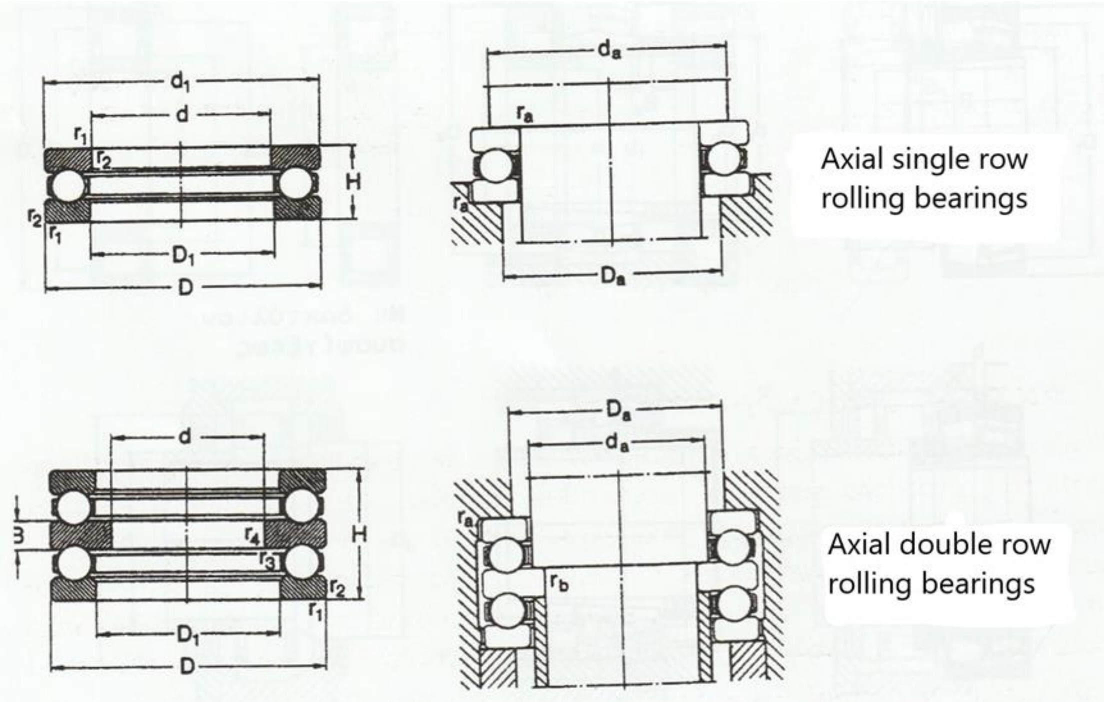
Self-regulating rolling bearings



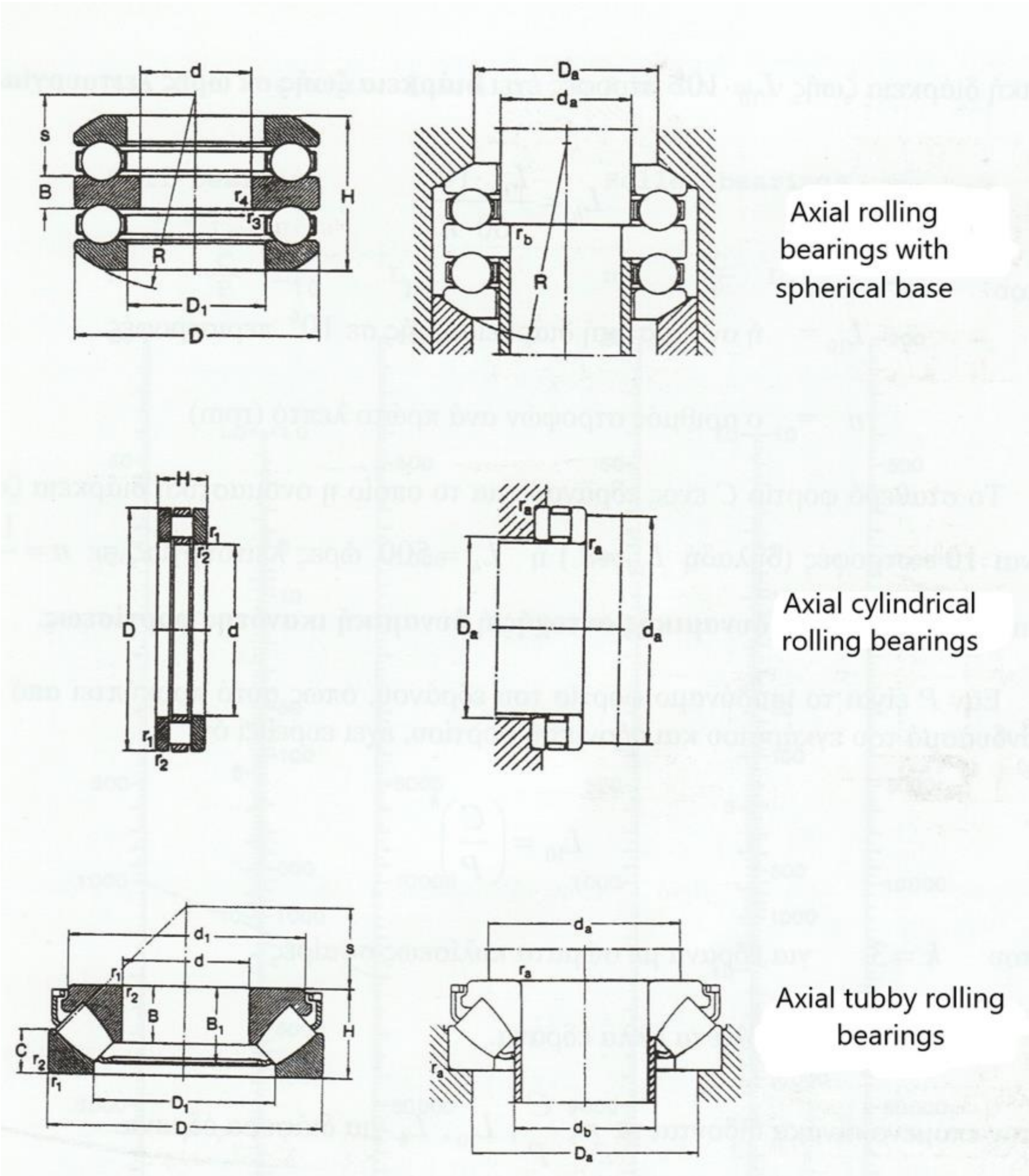
Sketch 3.6 Cylindrical rolling bearings



Tubby rolling bearings



Sketch 3.9a Axial inlet rolling bearings



Sketch 3.9b Axial rolling bearings

3.2 BEARING LIFE

The total number of rotations that a bearing can make, operating normally with its load, until signs of fatigue appear in some of its components is called the life of the bearing. When 10% of the large number of similar bearings (same dimensions and other characteristics, with the same loads and operating conditions) show signs of fatigue of a certain life span and then this number of rotations is called the nominal lifetime of the whole series of similar bearings to wit $L_{10} \cdot 10^6$ turns, in which the 90% of the bearing series will withstand. A bearing with nominal lifespan $L_{10} \cdot 10^6$ turns has a lifespan of operating hours

$$L_h = \frac{L_{10} \cdot 10^6}{60 \cdot n}$$

Where

L_{10} = nominal lifespan in 10^6 rotations

n = number of turns per first minute (rpm)

The constant load C of a bearing, for which the nominal lifespan is 10^6 turns (in wit $L_{10} = 1$) or $L_h = 500$ operating hours in $n = \frac{100}{3}$ rpm is called the number of dynamic strength or dynamic sufficient load.

If P is the equivalent load of the bearing, as it results from the combination of the transverse and axial load, is found

$$L_{10} = \left(\frac{C}{P}\right)^k$$

Where k = 3 for bearings with ball rolling bodies

$k = \frac{10}{3}$ for the rest of the bearings.

n, $\frac{C}{P}$, L_{10}, L_h for various bearings, are given in the following table.

If various factors affecting the actual lifespan of the bearing (wear, material, operating conditions) are taken into account then based on the new life expectancy theory it will be

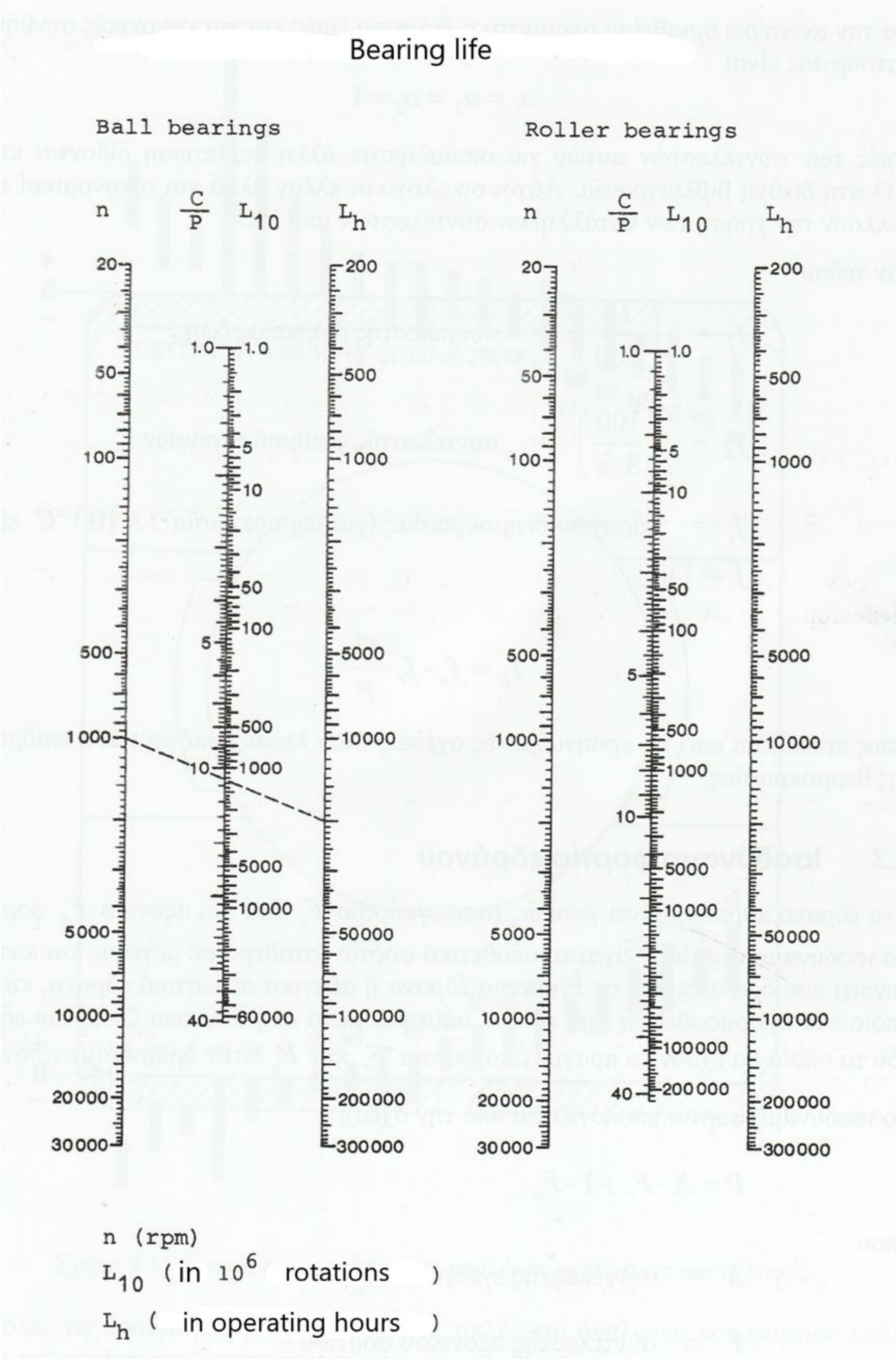
$$L = \alpha_1 \alpha_2 \alpha_3 L_{10}$$

Where

α_1 = confidence factor

α_2 = material factor

α_3 = operating conditions factor



Sketch 3.10 Bearing life

For the nominal lifespan and for normal operating conditions above, is

$\alpha_1 = \alpha_2 = \alpha_3 = 1$

Rates of these factors for any case are given by ISO in the international bibliography. Ecological and financial reasons impose the use of the appropriate factors by ISO.

If

$f_L = (\frac{L_h}{500})^{1/k}$ = lifespan factor

$f_n = (\frac{100}{3 \cdot n})^{1/k}$ = factor of number of turns

f_t = temperature factor (for temperature t<100ºC, $f_t = 1$)

Then

$f_L = f_n \cdot f_t \cdot \frac{C}{P}$

As it results from the previous relations when the effect of the temperature is also taken into account.

3.3 EQUIVALENT BEARING LOAD

A bearing generally receives both transverse and axial loads. The equivalent load P is the hypothetical load (constant in size and direction) acting radially on transverse bearings or axially on thrust bearings and if applied will have the same effects on the bearing life which will have real loads F_r and F_α when they act simultaneously.

The equivalent load is calculated by the relation

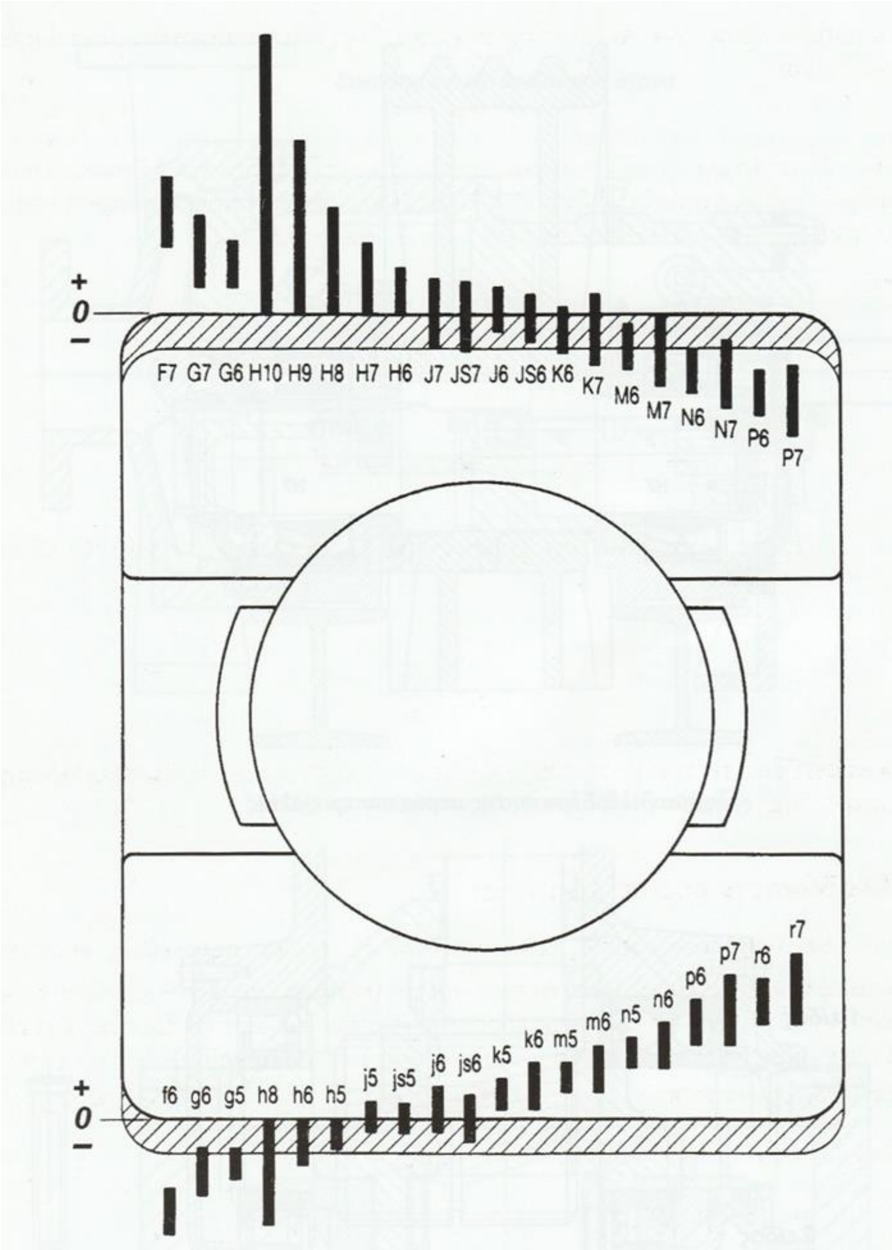
$$P = X \cdot F_r + Y \cdot F_\alpha$$

Where

X = coefficient of the transverse load

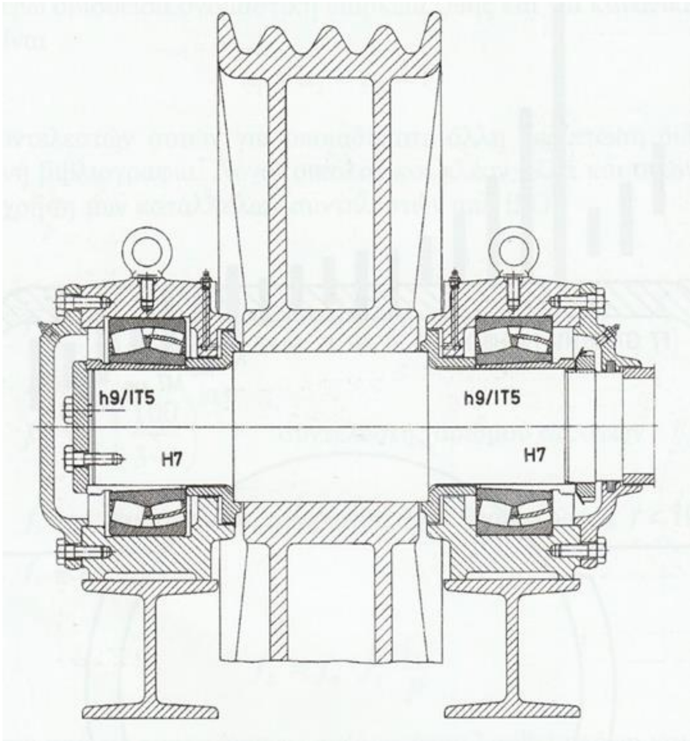
Y = coefficient of the axial load

X, Y coefficients are given in the bearing catalogues.

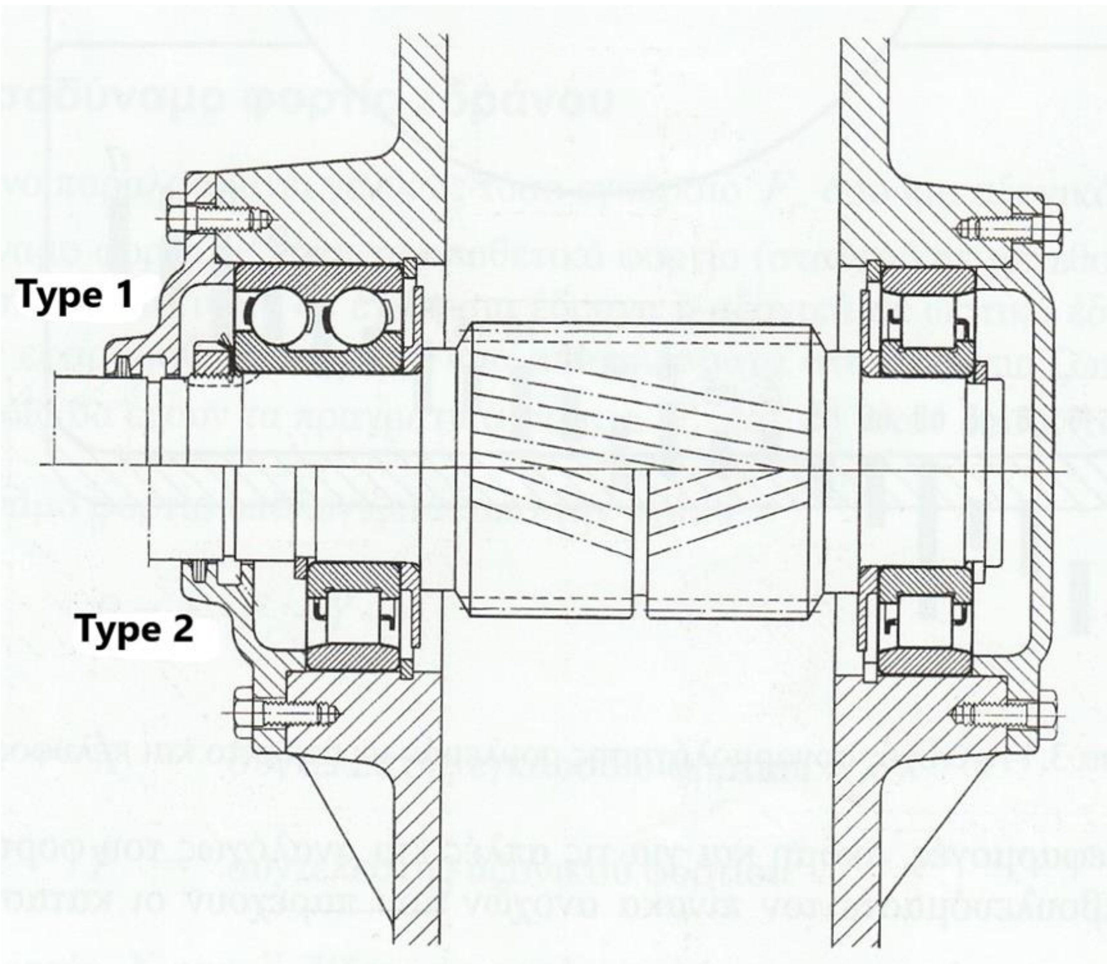


Sketch 3.11 Bearing assembling tolerances with shaft and hull

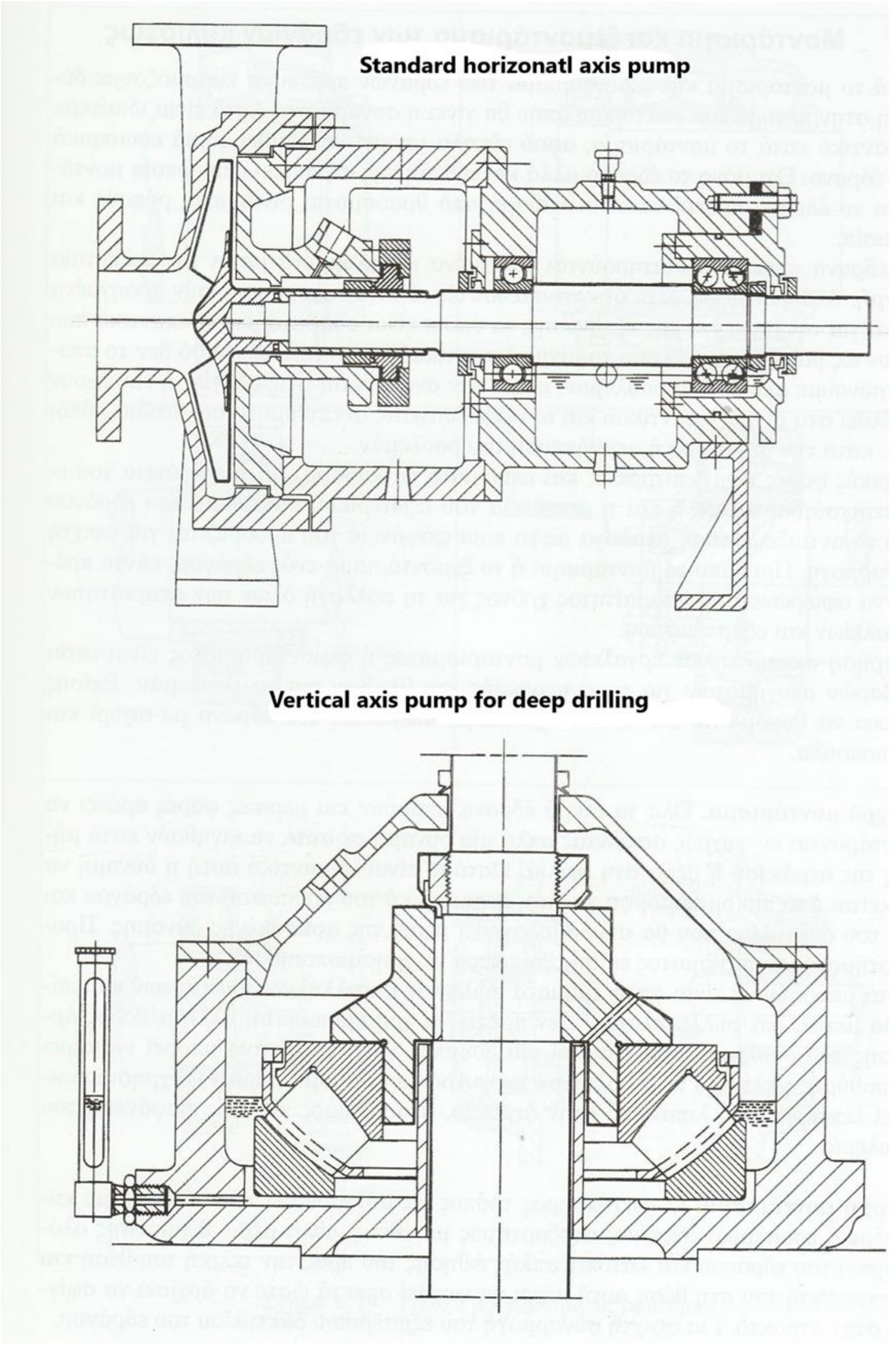
For every application, even the simple ones and depending on the load it is a good idea to consult the tolerance table provided by the bearing manufacturers.



Sketch 3.12 Pulley shaft bearing



Sketch 3.13 Typical shaft bearings



3.4 ASSEMBLY AND DISASSEMBLY OF THE ROLLING BEARINGS

When assembling and disassembling the bearings, we must apply force on the side of the bearing where the assembly will take place. This is especially important while assembling, as damage to the bearing can easily occur internally. Not only the bearing but also the position of the shaft in which the bearing is assembled must be clean of debris, chips, dirt and moisture.

Bearings should be retained, wrapped, or covered until the last moment. Since all bearing manufacturers use advanced materials to prevent oxidation, which are compatible with petroleum-based lubricants, the mixture of lubricant and antioxidant should not be removed from the new bearing, unless referred to in the instructions or unwanted particles, pollutants, etc. have entered the mixture of the lubricant and antioxidant, during transport or storage of the bearing.

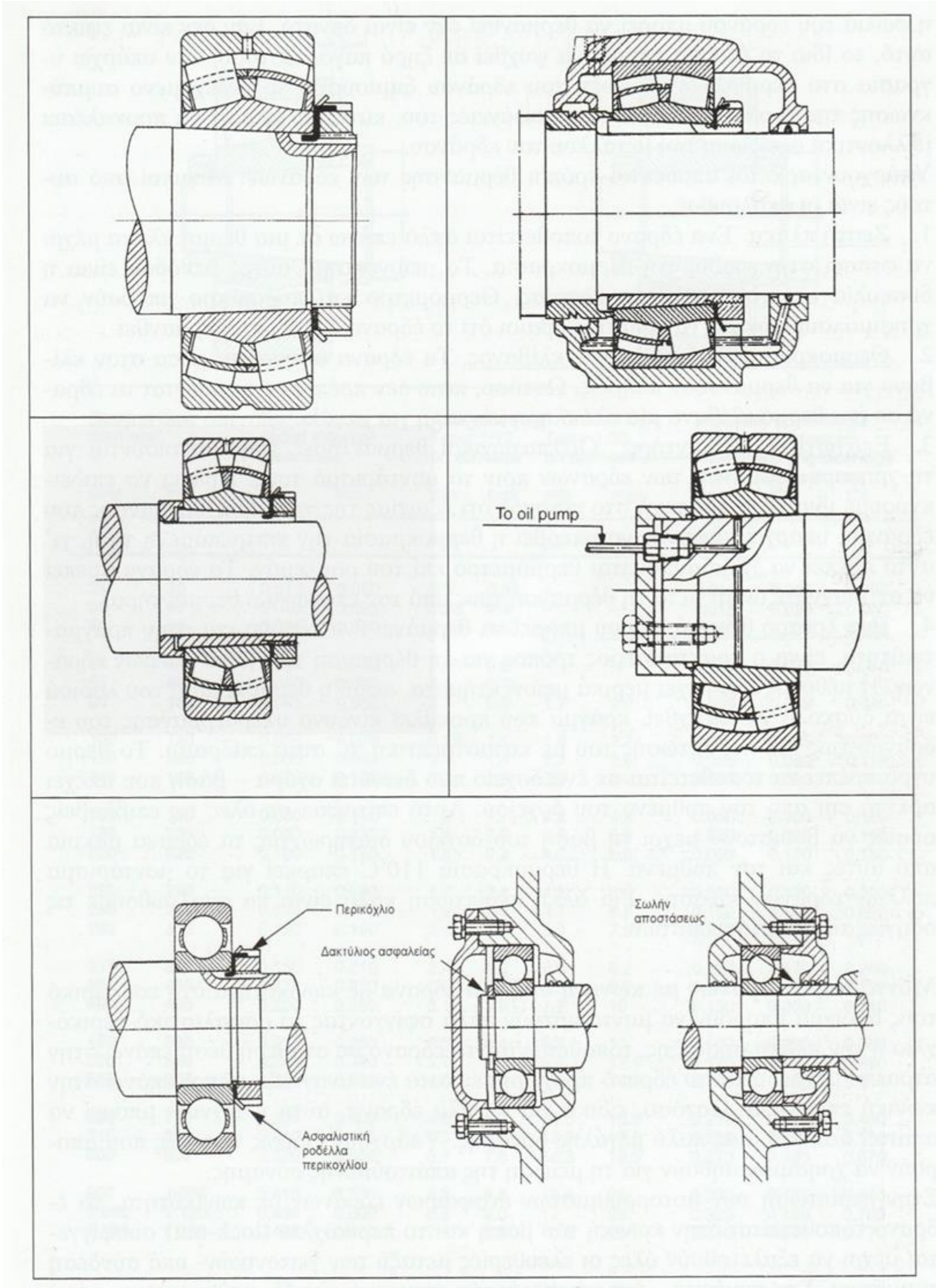
Sometimes it is permissible and desirable to clean the surface of the inner ring or the surface of the outer ring of the bearing from the antioxidant, depending on which surface it is adapted for a tight assembly. Before assembling or disassembling a bearing, the time required to collect all the necessary components must always be taken.

The use of unsuitable assembling or disassembling tools is the cause of serious accidents for technicians and damage to bearings. We must also remember that we never hit a bearing with a hammer or a sledgehammer directly.

Cold assembling. All small bearings can and sometimes have to be assembled in cold, simply exerting a force on them to move along the shaft or in the nest. However, it is important for this force to be exerted as evenly as possible, around the bearing and on the ring to be assembled through the force exerted. Assembling attachments should also be used.

These can be simple pipe sections of suitable dimensions and flat metal plates. No other bearing assembling method should be used, such as hammer, as the bearing may tilt to an undesirable-angle. For easily cold assembling, fine lubricant can be used on the shaft, the hull and the bearing surfaces.

Hot assembling. The easiest way to assemble each bearing with a cylindrical inner opening, regardless of size, is by heating the entire bearing and then simply pushing it to its final position and holding it in that position until it cools enough to start tightening to the shaft. For a tight assemble at the outer bearing ring,



Sketch 3.14 Ways of bearing assembly

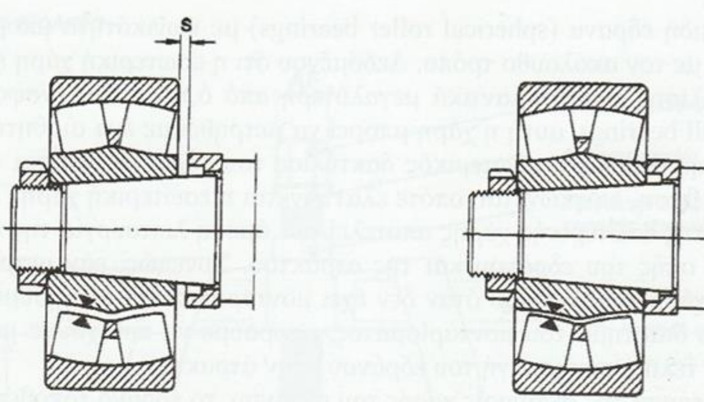
the bearing housing can be heated if possible. If this is not possible, the bearing itself can be cooled on dry ice. However, if there is moisture in the environment, the cooling of the bearing creates the possibility of moisture condensation on its surfaces, which may cause oxidation of the bearing metal in the future.

There are several acceptable ways to heat bearings: some of them are the following:

1. Hot plate. A bearing is simply placed on a hot plate until it reaches the desired temperature. The disadvantage of this method is the difficulty of controlling the temperature. Thermometer and pyrometer can be used to make sure that the bearing does not overheat.
2. Temperature controlled oven. The bearings are left in the oven to fully heat up. However, they should never be left in a hot oven overnight or for long periods of time.
3. Induction heaters. Induction heaters are used to quick heat bearings before assembling. We must pay special attention to the fact that due to the rapid heating of the bearing, there is a risk that the temperature will exceed the permitted value, so a thermometer must be used on the bearing. The bearings must be demagnetized after heating by the induction heater.
4. A hot oil bath can heat a bearing and is actually, the most practical way to heat larger bearings. This method has some disadvantages as the oil temperature is difficult to control, which causes the bearing to overheat or even glow in a destructive effect. The hot liquid should be placed in a container having a lattice well away from the bottom of the container. This allows all harmful substances to sink to the base of the container keeping bearings away from them and the bottom. The temperature of 100ºC is sufficient for assembling many roller bearings. For another case it is good to follow the instructions of the manufacturers.

Bearings with conical hole assembling. Conical bearing in their internal opening can be assembled simply by tightening the locking nut or clamping plate, placing the bearing in a fixed position on the shaft until the bearing extends by a necessary length over the conical surface. However, especially for large bearings, this technique may require the exercise of a very large force. There are special techniques that can be used to reduce the required force.

In the case of self-adjusting tapered bearings, the bearing is assembled on its conical base and the nut is tightened until all freedoms between adjacent-coupled sections are eliminated. Then, using a mechanical wrench, tighten the nut one-eighth of a turn further or as the catalogues show.



Bearing bore diameter d		Reduction in radial internal clearance		Axial drive-up Taper 1:12 on diameter		Taper 1:30 on diameter		Minimum permissible residual clearance[2)] after mounting bearings with initial clearance		
over	incl.	min	max	min	max	min	max	Normal	C3	C4
mm		mm		mm				mm		
24	30	0.015	0.020	0.3	0.35	–	–	0.015	0.020	0.035
30	40	0.020	0.025	0.35	0.4	–	–	0.015	0.025	0.040
40	50	0.025	0.030	0.4	0.45	–	–	0.020	0.030	0.050
50	65	0.030	0.040	0.45	0.6	–	–	0.025	0.035	0.055
65	80	0.040	0.050	0.6	0.75	–	–	0.025	0.040	0.070
80	100	0.045	0.060	0.7	0.9	1.7	2.2	0.035	0.050	0.080
100	120	0.050	0.070	0.75	1.1	1.9	2.7	0.050	0.065	0.100
120	140	0.065	0.090	1.1	1.4	2.7	3.5	0.055	0.080	0.110
140	160	0.075	0.100	1.2	1.6	3.0	4.0	0.055	0.090	0.130
160	180	0.080	0.110	1.3	1.7	3.2	4.2	0.060	0.100	0.150
180	200	0.090	0.130	1.4	2.0	3.5	5.0	0.070	0.100	0.160
200	225	0.100	0.140	1.6	2.2	4.0	5.5	0.080	0.120	0.180
225	250	0.110	0.150	1.7	2.4	4.2	6.0	0.090	0.130	0.200
250	280	0.120	0.170	1.9	2.7	4.7	6.7	0.100	0.140	0.220
280	315	0.130	0.190	2.0	3.0	5.0	7.5	0.110	0.150	0.240
315	355	0.150	0.210	2.4	3.3	6.0	8.2	0.120	0.170	0.260
355	400	0.170	0.230	2.6	3.6	6.5	9.0	0.130	0.190	0.290
400	450	0.200	0.260	3.1	4.0	7.7	10	0.130	0.200	0.310
450	500	0.210	0.280	3.3	4.4	8.2	11	0.160	0.230	0.350
500	560	0.240	0.320	3.7	5.0	9.2	12.5	0.170	0.250	0.360
560	630	0.260	0.350	4.0	5.4	10	13.5	0.200	0.290	0.410
630	710	0.300	0.400	4.6	6.2	11.5	15.5	0.210	0.310	0.450
710	800	0.340	0.450	5.3	7.0	13.3	17.5	0.230	0.350	0.510
800	900	0.370	0.500	5.7	7.8	14.3	19.5	0.270	0.390	0.570
900	1 000	0.410	0.550	6.3	8.5	15.8	21	0.300	0.430	0.640
1 000	1 120	0.450	0.600	6.8	9.0	17	23	0.320	0.480	0.700
1 120	1 250	0.490	0.650	7.4	9.8	18.5	25	0.340	0.540	0.770

Table 3.1 Assembling of barrel bearings with internal conicity S=axial drive-up

Barrel-shaped conical bearings can be assembled in the following way. Since the internal grace of such a roller bearing is significantly greater than that of a ball bearing, this grace can be measured with a measuring sensor. As the inner ring of the bearing is pushed onto its conical base, it swells, thus reducing the internal grace. The amount of internal grace reduction is a direct function of the joint between the bearing hole and the shaft. Therefore, if we measure the internal grace of the bearing when it is not assembled and control its reduction during the assembling, we can achieve with great accuracy the final assembly of the bearing on the shaft.

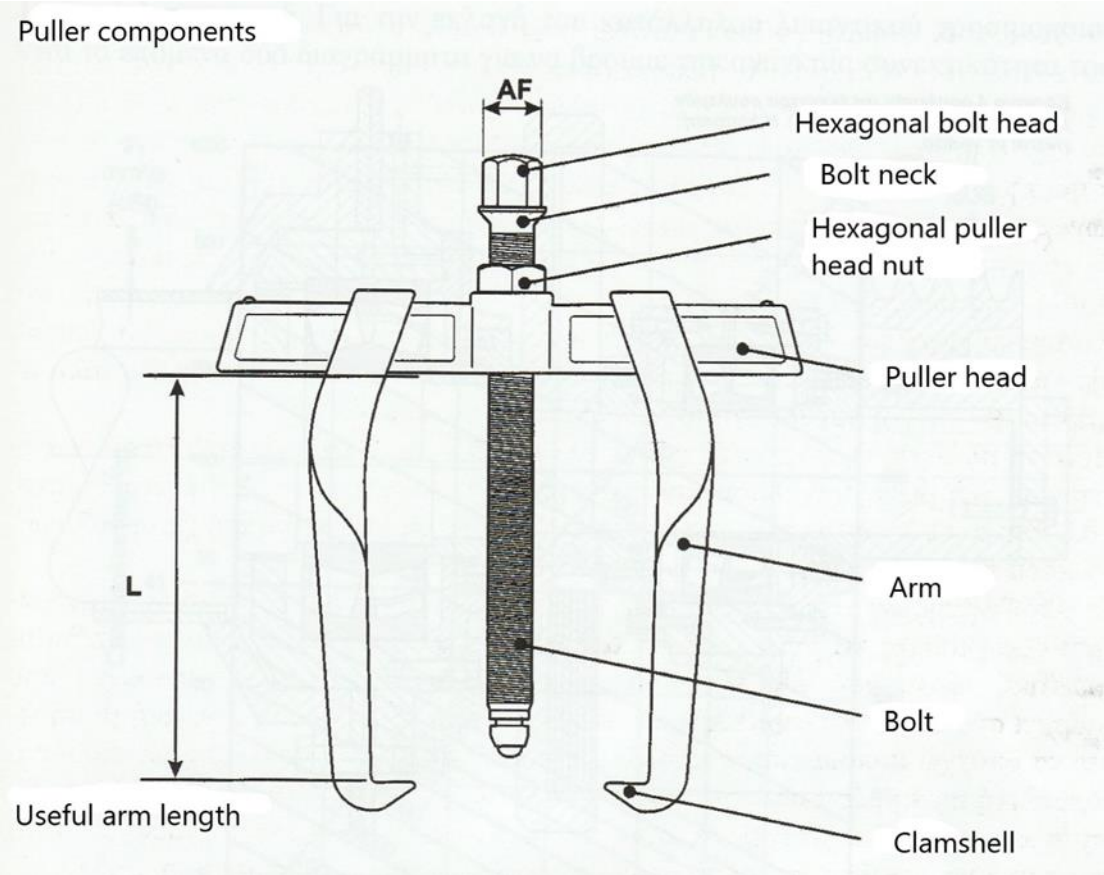
After measuring the radial grace of the bearing, the bearing is placed on its conical base. The locking nut can be fastened to the bearing, pushing it to move on the conical base until the inner grace is reduced by the size provided in table 3.1.

The size of the force required to position a bearing on the conical base can be reduced if the shaft is prepared to receive hydraulic pressure from a hydraulic pump through a suitable socket at the end of the shaft. We approach the bearing at the conical base so that there is some contact between them. Then we gradually develop hydraulic pressure through the bearing hole to inflate it. A pressure above 200 bar may be necessary for this purpose, but with this pressure exerted between the bearing hole and the shaft, it is easy to move the bearing on the conical surface of the shaft with much less torque to the lock nut or on the clamping plate in relation to simple assembling.

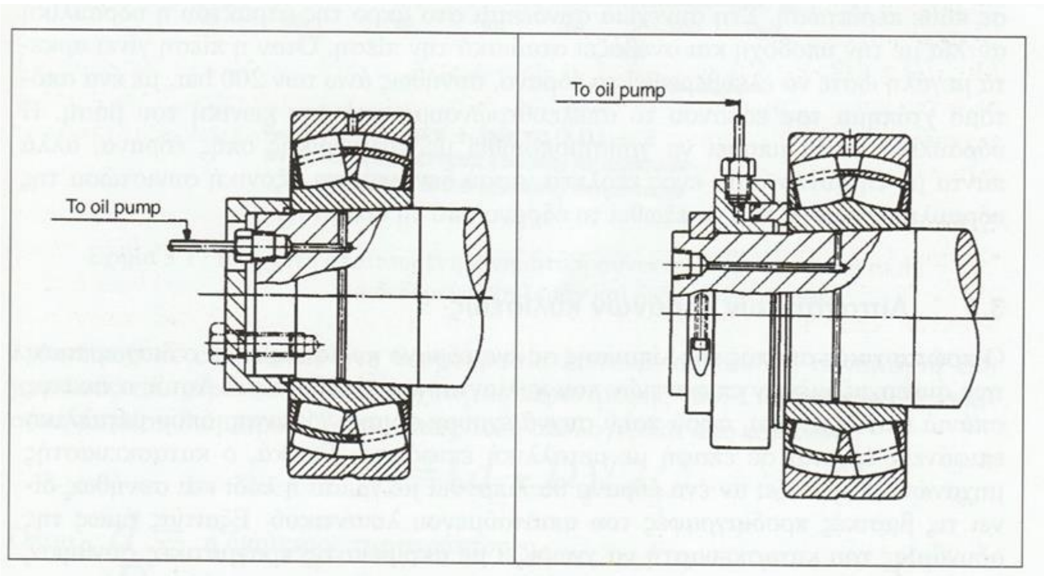
Another method for tapered bearings is to use a hydraulic nut or assembling tool as shown in the sketch. This technique can also be applied to the assembly of the cylindrical components when the dimensions of these and the shafts are large.

Disassembly of bearings. A wide variety of commercially available tools are designed to remove the roller bearings from place without damage. Bearing pulleys, sketch 3.15 are widespread. When disassembling the bearings, force is always applied to the ring that is pinned to the shaft or nest. Therefore, we must pay attention to which ring the puller will fit. Sometimes it is necessary to use other components besides the extractor.

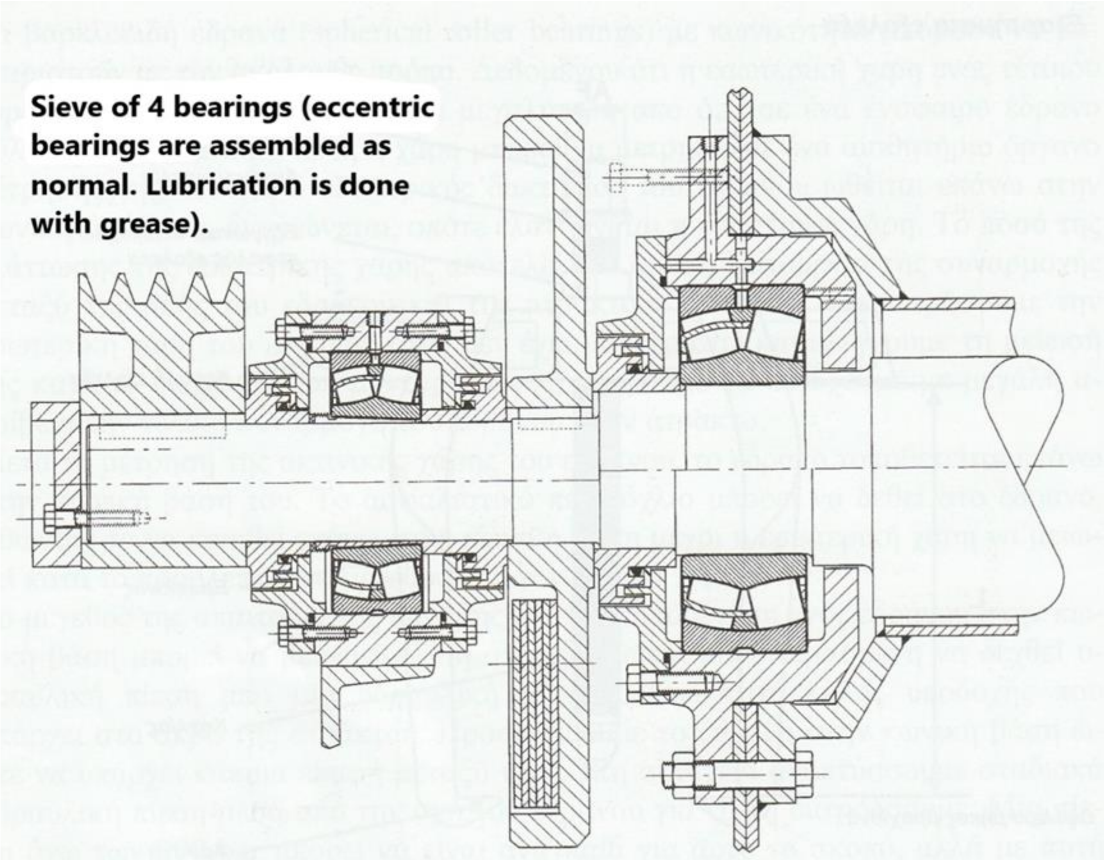
Use of hydraulic assistance in disassembly. The disassembly process is quite simple in the cases that the tractors are designed so that hydraulic pressure can be applied during the shaft and bearing connection process. First of all, the locking device of any shape must be moved for a while even in the axial movement of the bearing when it is released from the shaft. Few is enough in any case. Then the hydraulic pump is connected to the socket at the end of the shaft and gradually raises the pressure. When the pressure becomes high enough to release the bearing, usually over 200 bar with a sharp blow of the bearing we release it from conical base. The hydraulic pressure can be used with cylindrical bore bearings, but always with the cooperation of an extractor, since there is no axial component of the hydraulic pressure to push the bearing out of place.



Sketch 3.15 Mechanical roller bearing puller



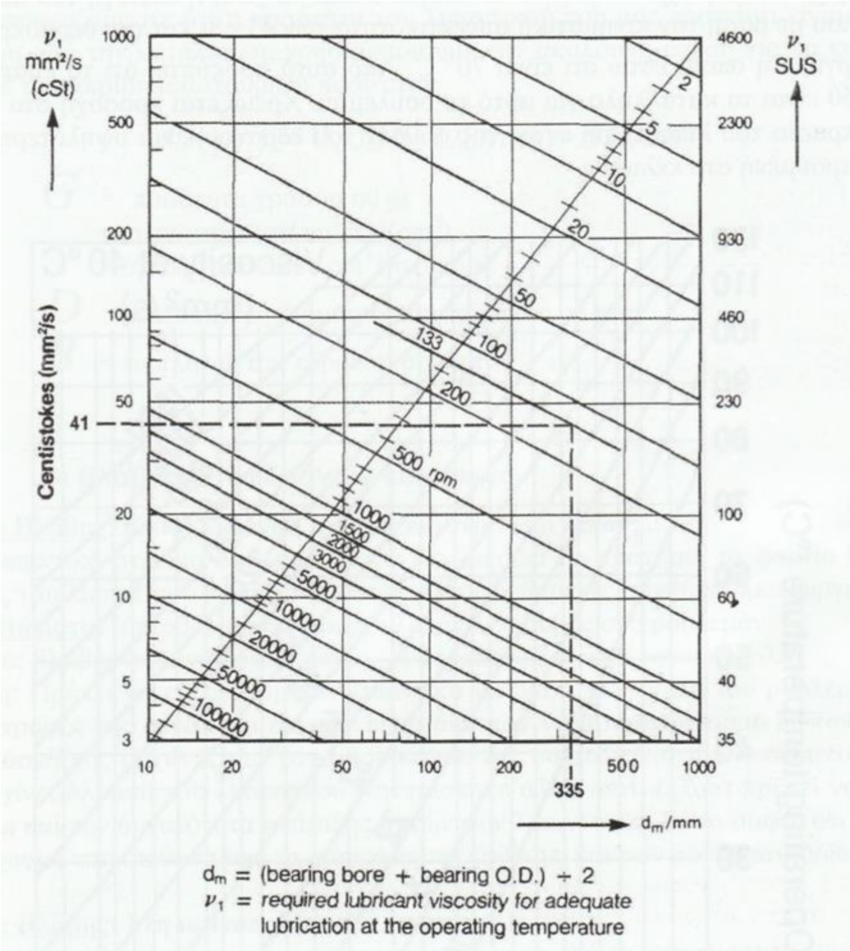
Sketch 3.16 Appliance of hydraulic power in bearings



3.5 LUBRICATION OF ROLLER BEARINGS

The primary purpose of lubrication on a roller bearing is to separate the cooperating surfaces from rolling and sliding. This goal is rarely achieved, since we have very often marginal lubrication, where a metal surface comes in contact with a metal surface. Generally, the engine manufacturer decides whether a bearing will be lubricated with grease or oil and usually gives the basic specifications of the required lubricant. However, due to the inability of the manufacturer to know accurately the actual operating conditions of the machine inside the industrial area, a study of the need for lubrication equipment is required by the maintenance staff.

Selection of the lubricant. The following diagrams are used for the selection of the suitable lubricant in order to find its necessary coherence



Sketch 3.17 Minimum required coherence of the lubricant for given bearing and speed

in centistokes in its operating temperature as a function of the size of the bearing and its rotation speed. For a roller bearing with average diameter 335mm, as it is calculated by the form

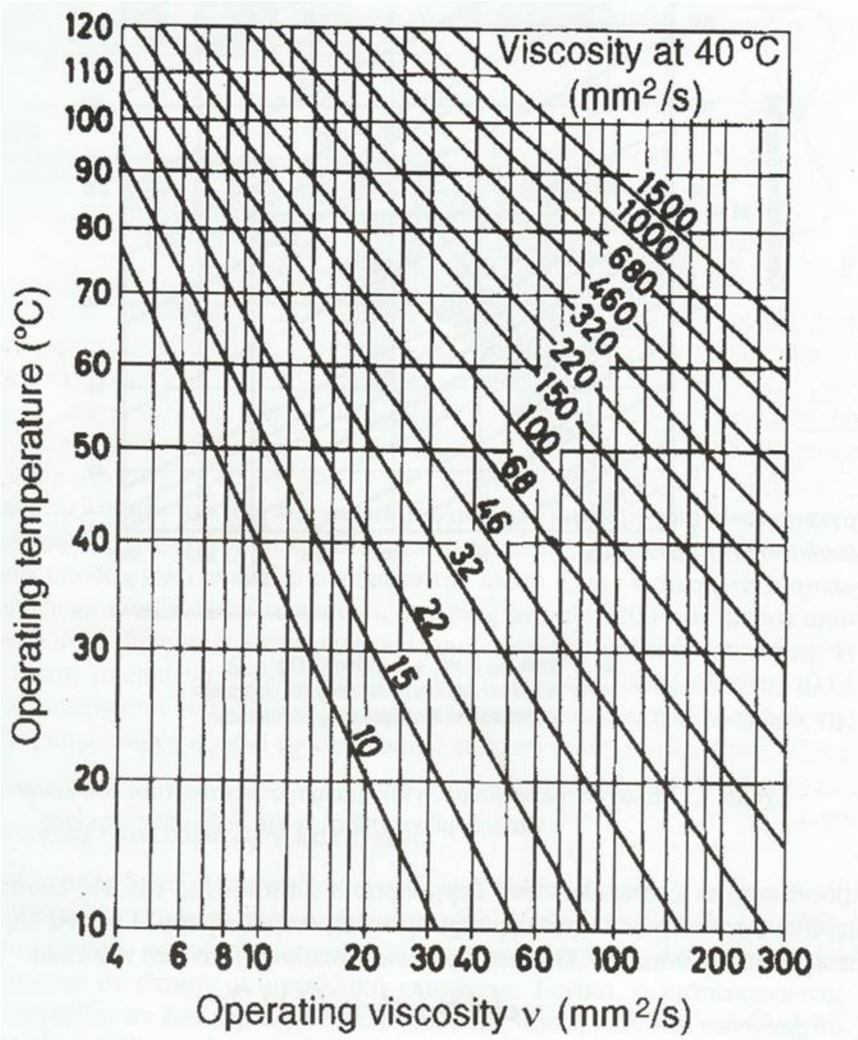
$$d_m = (d + D)/2$$

Where d = the diameter of the shaft

D = the outer diameter of the bearing

Which rotates at 133 rpm requires a lubricant with a kinematic coherence of 41 cSt at its operating temperature.

Sketch 3.18 contains the lubricants from which the most suitable will be selected based on the kinematic coherence of 41 cSt and the operating temperature which is at least 70ºC. As a result from the above the lubricant ISO 150 is the most suitable for this bearing. Attention is needed in that the temperature of the lubricant inside the bearing housing is higher than that measured in the hull.



Sketch 3.18 Kinematic coherence of lubricants

Grease replacement should only be done on the bearings. The amount of grease depends on the size of the bearing. If degreasing is available from the bearing manufacturer, we follow it. If not, or if we suspect that the amount of lubricant the manufacturer is proposing is greater than the appropriate one, we use the following relation to determine the exact quantity required:

$$G(gr) = 0{,}005 \cdot D(mm) \cdot B(mm)$$

Where G = amount of grease in gr

= density (gr/cm^3) ×V (cm^3)

= grease volume in cm^3 (approximately)

D = the outer diameter of the bearing in mm

B = the width of the bearing in mm

3.6 ROLLER BEARING FAILURE

Cause of failure: poor lubrication from an incorrect lubricant

The lubricant layer is broken and cannot hold loads because the type of the lubricant is not correct for the given operating conditions.

Symptoms: bearing overheating. Noise in the bearing. Frequent failures and replacements. The shaft does not rotate well.

Solution: we must examine the lubricant based on the bearings loads and shaft speeds and if possible consult with the machine manufacturer to determine the correct type of lubricant. If the lubricant has been changed (bought elsewhere) we must also check the possibility of mixing these lubricants. It is clear that we must have at our disposal the viscosity diagrams of both lubricants.

Cause of failure: unsatisfactory lubrication

Low level of lubricant-lubricant leaks from sealants.

Symptoms: bearing overheating. Noise in the bearing. Frequent failures and replacements. The shaft does not rotate well.

Solution: in lubrication with oil, the oil level must be slightly below the center of the lower roller bearing body. In grease lubrication, the available bearing space must be filled with grease by 1/3 to ½.

Cause of failure: lubricant excesses

The oil level in the bearings is too high or too much grease has been applied to the bearing. The large amount of lubricant (oil or grease) causes high circulation in the bearing resulting in increased bearing operating temperatures and high lubricant leakage.

Symptoms: overheating of the bearing. The shaft does not rotate easily.

Solution: in lubrication with grease, the amount of grease is removed up to the normal amount of grease in the bearing (1/3 to ½ of the space in the bearing). In lubrication with oil, the oil level should not exceed normal (slightly below the center of the lower roller bearing body).

Cause of failure: bearings with small internal radial grace.

Wrong choice of assemblies. The bearing has an unsuitable (small) internal grace for operating conditions, where the heat is transferred through the shaft to the inner ring of the bearing. This creates increased expansion in the inner ring, which cannot be covered by the small grace of the bearing.

Symptoms: noise in the bearing. Frequent failures and replacements. Low engine performance. The shaft does not rotate well.

Solution: a check is made to determine if the grace of the assembled bearing is that predicted by the design.

If not, then the bearing has been chosen incorrectly. We have to change the bearing with another one that has a greater radial grace. E.g., if the bearing is NORMAL we have to change it with C3.

If so, we need to check what happened and follow the assembly instructions.

Cause of failure: foreign substances act as abrasives on bearings

Sand, coal or other contaminants enter the inside of the bearing.

Symptoms: noise in the bearing. Frequent failures and replacements. Vibrations and shocks. Low engine performance. The shaft does not rotate well.

Solution: we need to clean the inside of the bearing, replace the damaged seals or improve the design of them to achieve proper sealing and protection of the bearing.

Cause of failure: foreign substances that act as corrosives on bearings

Water, acids, paints or other corrosive substances enter the inside of the bearing.

Symptoms: noise in the bearing. Frequent failures and replacements. Vibrations and shocks. Low engine performance. The shaft does not rotate well.

Solution: we must install a guard (shield or flinger) to prevent foreign substances from entering the bearing.

Cause of failure: the bearing is crushed (wedged) in the hull

The diameter of the hull (hole) is small, the side surface of the hull is not cylindrical, and it is distorted or deformed. The side bearing surface of the bearing base in the construction is not flat.

Symptoms: noise in the bearing. Frequent failures and replacements. Vibrations and shocks. Low engine performance. The shaft does not rotate well.

Solution: smooth opening of the hull to release the wedged bearing. Check the side assembling position of the bearing base so that it is flat and that the adjusting shims cover the entire bearing base. Pay attention to the diameter of the shell when changing the bearing of the free bearing from cylindrical to deep groove.

Cause of failure: foreign substances in the hull of the bearing

Chips, turning waste or particles left in the shell and not removed before bearing assembly.

Symptoms: noise in the bearing. Frequent failures and replacements. Vibrations and shocks. Low engine performance. The shaft does not rotate well.

Solution: careful cleaning and removal of foreign substances must be done and new lubrication must be done.

Cause of failure: air falls on the bearing at high speed

When air falls on the bearings at high speed it creates different pressures, which causes a lubricant to leak.

Symptoms: frequent failures and replacements.

Solution: we place reflective plates to change the direction of the air. We must avoid creating different pressures on the bearings. If the problem still exists it is better to use grease as a lubricant.

Cause of failure: very tight sealants

Deformed sealants. The seals that touch the shaft (friction seals) dry or are subjected to high spring pressures.

Symptoms: overheating of the bearing. The shaft does not rotate well.

Solution: we have to replace the friction seals with others, the spring pressure of which will be correct or predictable. Sealants need to be lubricated so that they do not dry.

Cause of failure: improperly aligned seals, insufficient due to the labyrinth type seals, incorrect placement of the reflecting discs (flinger).

Rotating sealants or rotating particle reflecting discs are rubbed on fixed surfaces of the machine.

Symptoms: overheating of the bearing. Noise in the bearing. Frequent failures and replacements. The shaft does not rotate well.

Solution: we must control the grace of the particle seals and reflectors to avoid abrasive surfaces. The alignment of the components must be corrected.

Cause of failure: blocked lubrication oil return passages

Clogged-blogged oil return passages do not allow the pumped lubricating oil to leave the lubrication area and return to the lubrication tank, resulting in increased leakage.

Symptoms: overheating of the bearing.

Solution: clean the oil diodes. Remove the used oil and refill to the permitted level with fresh lubricating oil.

Cause of failure: cross-bearing with preload on cross support

Cross-bearing support of the fixed and free bearing of the shaft when the bearing distance is not short.

Symptoms: overheating of the bearing. Noise in the bearing. Frequent failures and replacements. Low engine performance. The shaft does not rotate well.

Solution: place metal blades between the lid and the hull of the bearing to release the axial load and the displacement of the bearing. The axial expansion of the shaft must not be prevented.

Cause of failure: preload bearings when the shaft has two fixed bearings

When the shaft has two fixed bearings its free expansion is prevented and the internal axial grooves of the bearings are usually insufficient to receive the necessary axial expansion of the shaft.

Symptoms: overheating of the bearing. Noise in the bearing. Frequent failures and replacements. Low engine performance. The shaft does not rotate well.

Solution: we have to remove the lid of one of the two bearings outwards. We place metal blades between the hull and the lid to create the appropriate grace. We can use an axial spring on the outer bearing ring to reduce the axial play of the shaft.

Cause of failure: loose bearings on the shaft

The diameter of the shaft is small. The bearing adapter sleeve does not have the proper tightening on the shaft.

Solution: we fill the shaft with metal and smooth it to achieve a suitable diameter and good tolerance. We reposition the adapter sleeve in order to achieve a stable fit of the bearing on the shaft.

Symptoms: noise in the bearing. Frequent failures and replacements. Vibrations and shocks. Low engine performance.

Cause of failure: bearing very tight on the shaft

Very tight fit the adapter sleeve of the bearing has a high tightness on the shaft.

Symptoms: overheating of the bearing. Noise in the bearing. Frequent failures and replacements. Low engine performance. The shaft does not rotate well.

Solution: loosen the nut and the bearing adapter sleeve. The bearing must rotate freely.

Cause of failure: freely rotating outer ring of the bearing

The hull diameter is large for the outer diameter of the bearing. The bearing is heated. The loads are unbalanced.

Symptoms: overheating of the bearing. Noise in the bearing. Frequent failures and replacements. Vibrations and shocks. Low engine performance.

Solution: it is necessary to balance the rotating masses. If necessary, replace the hull.

Cause of failure: noisy bearing

Flattened rolling body due to the sliding of the rolling body that occurs during rapid starts.

Symptoms: noise in the bearing. Vibrations and shocks. Low engine performance.

Solution: we check the rolling bodies if they have such a problem we replace the damaged scroll body through the inlet of the scroll bodies, if it exists. We must control the minimum required operating load of the bearings.

Cause of failure: bearing assembling on the shaft on a conical base. Bearing assembling on the hull in a conical hole

Fitting of concentrated load to the bearing. Poor load distribution on the bearing due to poor bearing geometry of the bearing shaft or hull.

Symptoms: noise in the bearing. Vibrations and shocks. Low engine performance.

Solution: we check the shaft and the hull and repair the wrong position to ensure correct geometry (shape and size). Repair may not be possible and the shaft or hull mat need to be replaced.

Cause of failure: micro gradient high of the shaft

Inappropriate gradation height of the shaft. The shaft is bent by a load that is large for the small gradation of the position.

Symptoms: Noise in the bearing. Frequent failures and replacements. Vibrations. Low engine performance. The shaft does not rotate well.

Solution: we process the shaft in this position to reduce the stress concentration. A sleeve may need to be fitted to the shaft. In any case we measure the diameters of the new gradation based on the bearing installation plans.

Cause of failure: the shaft gradation is too large

The gradient is contacted with the bearing seals. The large diameter of the shaft gradient in the bearing seals is rubbed.

Symptoms: overheating of the bearing. Noise in the bearing. Frequent failures and replacements. The shaft does not rotate well.

Solution: we process the shaft gradation to avoid contact with the bearing seals. We check the new dimensions according to the bearing assembly plans.

Cause of failure: the inside gradation of the hull is small

The height of the inside of the hull gradient is small and is not enough to support the bearing. Improper alignment of this hull leads to distortion of the outer bearing ring.

Solution: we process the hull in this gradation to reduce the stress concentration. A sleeve may need to be fitted to the hull. In any case we check the diameters of the new gradation based on the bearing installation plans.

Cause of failure: the inside gradation of the hull is large

The bearing seals are deformed.

Symptoms: Noise in the bearing. Frequent failures and replacements. The shaft does not rotate well.

Solution: We process the gradation of the hull to adjust the geometry and release the seals.

Cause of failure: the radius of curvature of the shaft gradation is large

The Bearing cannot fit properly in its position on the shaft. The shaft and the inner ring of the bearing are deformed.

Symptoms: Noise in the bearing. Frequent failures and replacements. Vibrations. Low engine performance. The shaft does not rotate well.

Solution: we process the shaft to reduce the gradient radius and achieve a good placement of the bearing in position on the hull.

Cause of failure: the radius of curvature of the inward of the hull is large

The bearing cannot fit properly in place on the hull. The gradation of the hull and the outer bearing ring are deformed.

Symptoms: Noise in the bearing. Frequent failures and replacements. Vibrations and shocks. Low engine performance. The shaft does not rotate well.

Solutions: we process the gradation of the hull to reduce the gradient radius and achieve a good placement of the bearing in position on the hull.

Cause of failure: poor (linear and angular) shaft alignment

Incorrect linear or angular alignment of shafts develops forces and creates problems in the operation of the bearings.

Symptoms: overheating of the bearing. Frequent failures and replacements. Vibrations and shocks. Low engine performance. The shaft does not rotate well.

Solution: we correct the alignment of the shafts using full depth metal blades. The axis of the shafts must be in line (on the same straight line).

Cause of failure: the nut lock is attached to the bearing

Symptoms: overheating of the bearing. Noise in the bearing. Low engine performance. The shaft does not rotate well.

Solution: remove the nut and straighten the teeth of the lock or replace it with a new one.

Cause of failure: deformed rolling body on the bearing

Any deformation in a roller bearing body is a cause for immediate replacement through the corresponding bearing slot. If it does not exist, the bearing cannot be used. These deformations can be caused by a sharp blow to the bearing. The bearing has been installed incorrectly, possibly with a hammer blow.

Symptoms: Noise in the bearing. Frequent failures and replacements. Vibrations and shocks. Low engine performance.

Solution: we replace the bearing with a new one. When assembling, never hit any part of the bearing with a hammer. We always use assembly tools.

Cause of failure: bearing noise from other components

The moving parts of a machine affect the behavior, operation and response of bearings.

Symptoms: Noise in the bearing.

Solution: we carefully check every moving part in the machine and if possible we isolate those that affect the operation of others.

Cause of failure: lubricant leakage and dirt enter in the bearing

Worn bearing seals. The seals attached to the shaft wear out excessively to allow the lubricant to leave the bearing or to allow external contaminants to enter the bearing.

Symptoms: overheating of the bearing. Noise in the bearing. Frequent failures and replacements.

Solution: we must replace the sealants in each degreasing of the bearing (removal of old lubricant and installation of new one).

Cause of failure: vibrations and shocks in the bearing due to great grace.

The excessive grace of the bearing creates vibrations.

Symptoms: noise produced from the bearing. Vibrations and shocks. Low engine performance.

Solution: we must use bearings with the appropriate internal grace. It is advisable to use a spring on the outer ring of the free-bearing bearing to reduce both the axial and radial play of the bearing.

Cause of failure: vibrations and socks to the bearings due to unbalance

A cam load and unbalance create vibrations. Then the machine has vibrations.

Symptoms: Noise in the bearing. Vibrations and shocks. Low engine performance.

Solutions: balance the rotating masses of the machine.

Cause of failure: bearing color change

The bearing was rather dissembled with flames, so the bearing changed color and became distorted. Distortion of the shaft and other components may occur due to local overheating.

Symptoms: noise in the bearing.

Solution: when assembling bearings, concentrated local overheating must be avoided anywhere as it may cause distortion. We have to replace the bearings that have changed color or that have been distorted.

Cause of failure: large shaft on the bearing

When the diameter of the shaft, on which the bearing is placed, is larger than it should be, then a large expansion of the inner ring of the bearing is caused. This reduces the available grace of the bearing.

Symptoms: overheating of the bearing. Noise in the bearing. Frequent failures and replacements. Low engine performance. The shaft does not rotate well.

Solution: we process the shaft to ensure a proper fit with the inner bearing ring. If you cannot process the shaft, replace the bearing with another that will have greater radial grace.

Cause of failure: enlargement on the hull diameter

The removal of small pieces from the soft material of the hull leads to an increase in the diameter of the hull, with the result that the outer ring bearing rotates freely inside the hull.

Symptoms: overheating of the bearing. Noise in the bearing. Frequent failures and replacements. Vibrations and shocks. Low engine performance.

Solution: we process the sleeve to the dimensions we want.

Cause of failure: bearings on stand-by machine

As the machine is in stand-by mode, it is exposed to the vibrations of neighboring machines. Thus causing the false brinelling effect on its bearings.

Symptoms: we carefully examine the bearing for wear points at the bearing positions of the rolling bodies. In stand-by machines the bearings withstand better the vibrations of neighboring machines. It is good to operate the stand-by machine for a while at regular intervals in order to avoid this phenomenon.

<u>Ten points that need attention in the bearing to succeed in the long run and without problems their operation:</u>

1. Careful handling of bearings.
2. Shaft and bearing inspection.
3. Avoid overheating.
4. Use the appropriate tools.
5. Replace the old one (if it was proper) with a same bearing.
6. Monitor the bearing-shaft assembly.
7. Do not clean new bearings (unless required).
8. Use the appropriate lubricant (schedule).
9. Rotate non-operating machines.
10. Investigating for dangerous signals.

The ten dangers of bearing reliability:

1. Wrong lubricant.
2. Mixing non-miscible lubricants.
3. More or less lubricant.
4. Pollutants or oxidants in the lubricant.
5. Poor component alignment.
6. Deformed hull.
7. Unsatisfactory internal use.
8. Problematic bearings of the shaft.
9. Deformed sealants or protectors.
10. Other causes (low C, angular deviation of the axis, etc.).

CHAPTER 4: COGWHEELS

4.1 GENERALLY

Gearwheels are mechanical elements that are used for rotational motion, transmission and power transfer from an engine to a moving shaft, through a successive involution of cooperating teeth. Teeth are formations of successive recesses and protrusions in the circumference of a cogwheel, such that it is possible for the recess of one wheel to cooperate with the protrusion of the cooperating cogwheel. They are made of different profiles, such as advanced, cycloid (orthocycloid, epicycloid, pericycloid) arc of circle, spiral, etc. The most common among them is the profile of the advanced, which is the curved one that defines the base of a curve.

On most types of wheels, the teeth are not theoretically necessary for the transmission of rotational motion. However, the need to transfer high torque and constant motion (i.e., constant gear relations) from the engine to the moving shaft necessitates the existence of teeth.

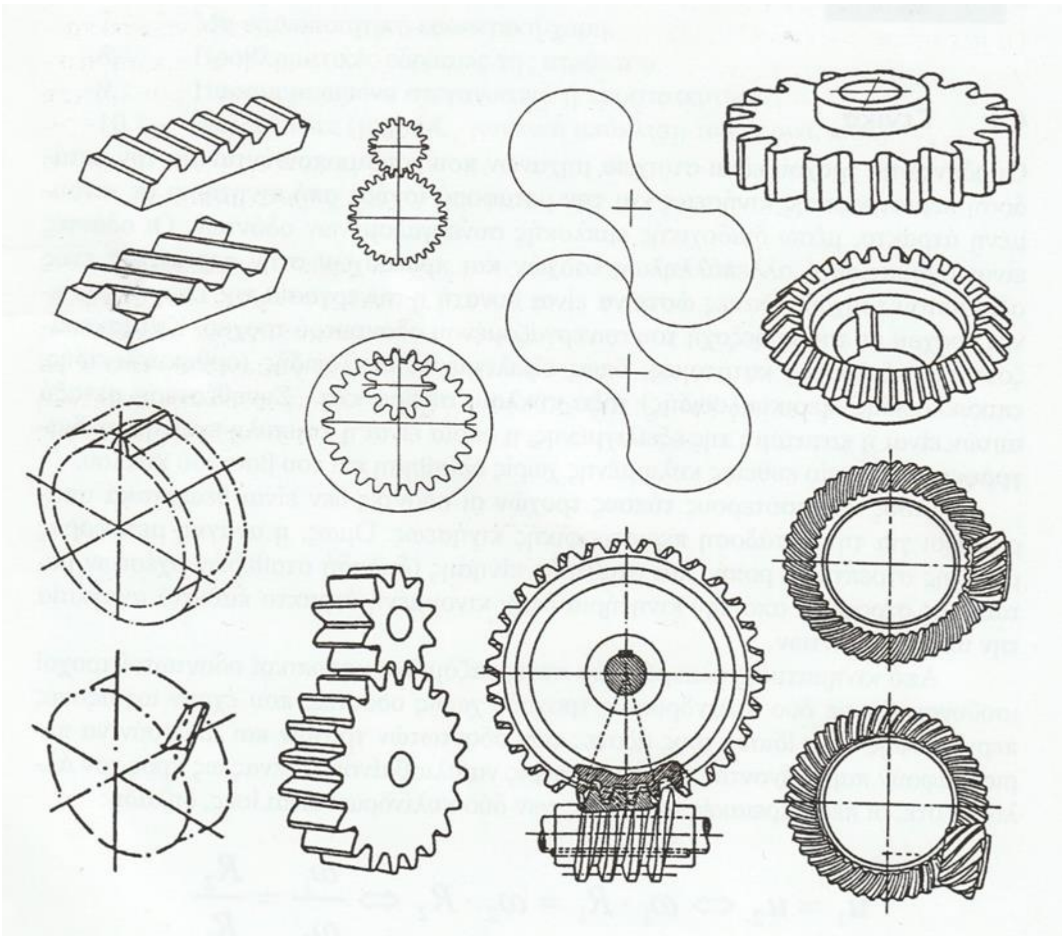
From a kinematic point of view, two cooperating front cogwheels are equivalent to two cylindrical gears without teeth, which have as rotation axes the same axes of the sprockets and can rotate while remaining in contact without sliding one towards the other. Then, the regional speeds of the two cylinders are equal, i.e.:

$$u_1 = u_2 \Leftrightarrow \omega_1 \cdot R_1 = \omega_2 \cdot R_2 \Leftrightarrow \frac{\omega_1}{\omega_2} = \frac{R_2}{R_1}$$

From which we understand that the transmission ratio of the motion is inversely proportional to the rays of the cylinders. Therefore, the rays of the rolling surfaces are an element of great importance in the cogwheels. In a front gear, the section of this cylindrical surface with the plane of rotation constitutes the initial cycle and thus each gear is characterized, among others, by the radius of the initial cycle. In abnormal gears such as the displaced ones, their rolling cycles are defined, with the beginning of the cooperation of the wheels.

4.2 TYPES OF COGWHEELS

The axis around which two cooperating sprockets rotate can be: parallel at a distance a, intersecting at an angle d or incompatible at a distance a, and an angle d.



Sketch 4.1 Types of cogwheels.

The relative position of these axles is crucial to the shape of the wheels to be used. The types of cogwheels are:

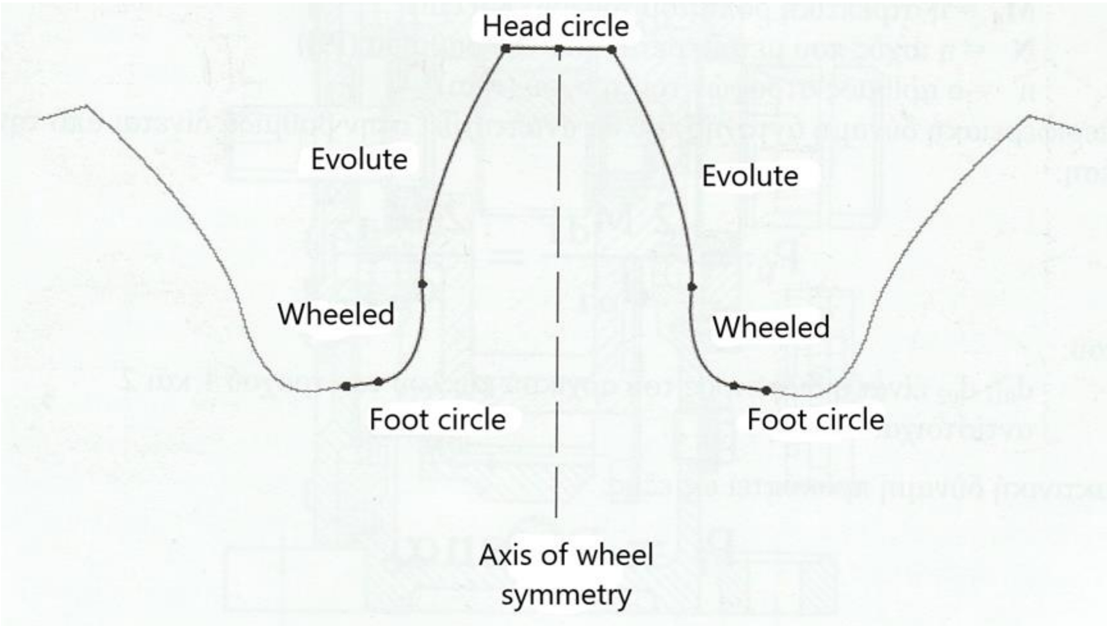
- Frontal (with straight or helical teeth).
- Conical.
- Helical incompatible axes.
- Endless screw-crown system.

We briefly list the characteristics of these cogwheels.

4.2.1 Front cogwheels

The front gears have a cylindrical shape and are distinguished into wheels with straight teeth and wheels with helical teeth. In the first case the teeth are parallel to the axis of the wheel. In the second case the teeth are inclined to the axis of the wheel, forming a constant angle with it, while in order to be able to cooperate, one must have right-handed and the other left-handed teeth.

The frontal straight teeth wheels have a degree of overlap of 1 to 2 while they are not suitable for large loads because they cause noise and create large impacts. In contrast, frontal helical teeth have a greater degree of overlap, but the disadvantage is that the presence of the angled propeller in the dentition results in the development of an axial load on the teeth, which stresses the fixed bearings.



Sketch 4.2 Profile of evolute straight gear teeth.

Front gears can have external serrations, i.e., the teeth are on the outer surface of the wheel or internal serrations, i.e., the teeth are on the inner surface of the wheel. In the case of a step, when two front wheels of different sizes are involved, the larger one is called a crown and the smaller one a pinion. In a simple reduction gear stage the incoming power and force are applied to the drive wheel, to the pinion. The power and force exiting the stage are transferred by the cooperating or moving wheel.

The drive wheel rotates the driven without slipping. Unlike a stepping wheel, the drive wheel is considered to be the largest. Sketch 4.2 shows the shape of a profile of a tooth of a straight frontal dentition by advanced wheel.

If the force to be transferred (through the step) is N and n_1 the turnings (per unit time) of the pinion, then the torque on the pinion will be:

$$M_{d1} = \frac{N}{\omega_1}$$

Where $\omega_1 = 2\pi n_1$ is the angle speed of the pinion.

The form below is successfully used for the calculation of the torque, for minus losses:

$$M_d = 71620\frac{N}{n}$$

M_d = torque of the wheel (Kp cm)

N = the power transferred from the step (PS)

n = the number of turns of the wheel (rpm)

The regional force strength developing in the step is given from the relation:

$P_u = \frac{2M_{d1}}{d_{01}} = \frac{2M_{d2}}{d_{02}}$

Where:

d_{01}, d_{02} is the diameter of the initial circle of the wheel 1 and 2.

The radial force results as follows:

$$P_r = P_u tana_0$$

The total force exerted on the wheel is given by the relation:

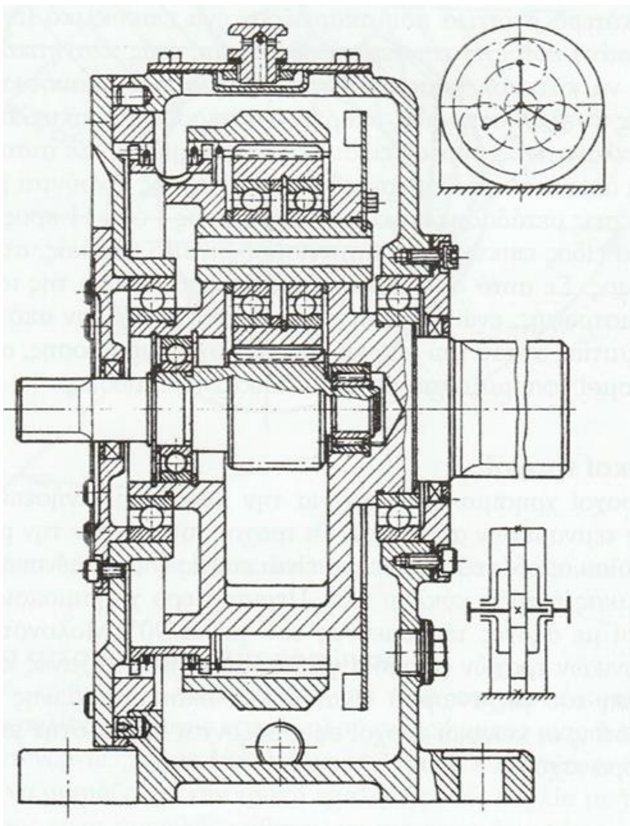
$$P = \sqrt{P_u^2 + P_r^2}$$

And by replacing the relations for P_r and P_u we have:

$$P = \sqrt{(\frac{2\, M_{d_1}}{d_{01}})^2 + (\frac{2\, M_{d_1}}{d_{01}}\ tana_0)^2} = \sqrt{(\frac{2\, M_{d_2}}{d_{02}})^2 + (\frac{2\, M_{d_2}}{d_{02}}\ tana_0)^2}$$

As we understand from the above load analysis for the front toothed wheels, the best signal definition (hence the better position of the sensor for impact pulse measurements) is achieved in the radial direction, since the component of the axial force is equal to zero (in contrast with helical front teeth wheels). The axial force of the helical wheels depends on the angle of the tooth propeller β_0 to the wheel axis and is calculated as:

$$P_a = P_u \cdot \varepsilon\varphi\beta_0$$



Application of frontal gears are planetary systems with their epicyclical mechanisms for optimizing the volume of the gearboxes, as in sketch 4.3.

In a cyclic driving system, the power is transmitted between the drive and the driving machine through multiple paths. The term epicyclical refers to the family of transmission systems, where one or more gears move coaxial gears during their circumferential motion, which can be fixed or rotate around their axis.

The gears that work in a circular mechanism can have straight serrations, helical or even double helical. Due to the existence of many modes of power transfer, an epicyclical mechanism is the ideal space solution for the transfer of a given power since it occupies a small volume. Other advantages of epicyclical mechanisms are the high degree of efficiency, the low inertia for the given transmitted power and the possibility of transferring high torque and power.

The basic elements that make up an epicyclical mechanism are the sun, a corona with internal indentation, a planetary carrier and the planets. There is an asteroid epicyclical mechanism, in which the planetary carrier is stationary, in order to combat the weakness of planetary systems in high-speed applications. In that type of system, the power input and output are moved in different directions, while the gear relations vary from 2 to 1 to 11 to 1.

Another type of epicyclical mechanism is the solar, in which the sun is stationary. Here the input and output shafts have the same rotation time, while the gear ratios vary from 1.1 to 1, to 1.7 to 1. Due to this small range of transmission ratios, the solar epicyclical mechanisms are applied to many special cases.

4.2.2 Conical wheels

Conical wheels are used to transmit motion and transfer power between intersecting shafts. These wheels are in the form of a truncated cone whose teeth can be straight or inclined as advanced or helical parts or circular arches etc. The most commonly used are conical wheels with axes intersecting at the angle of 90º. Even though, the conical wheels construction (which is done on special cutting machines) as the assembly of the mechanism is more complicated than that of the front wheels, yet the conical wheels are often applied in the transmission of motion and power transfer.

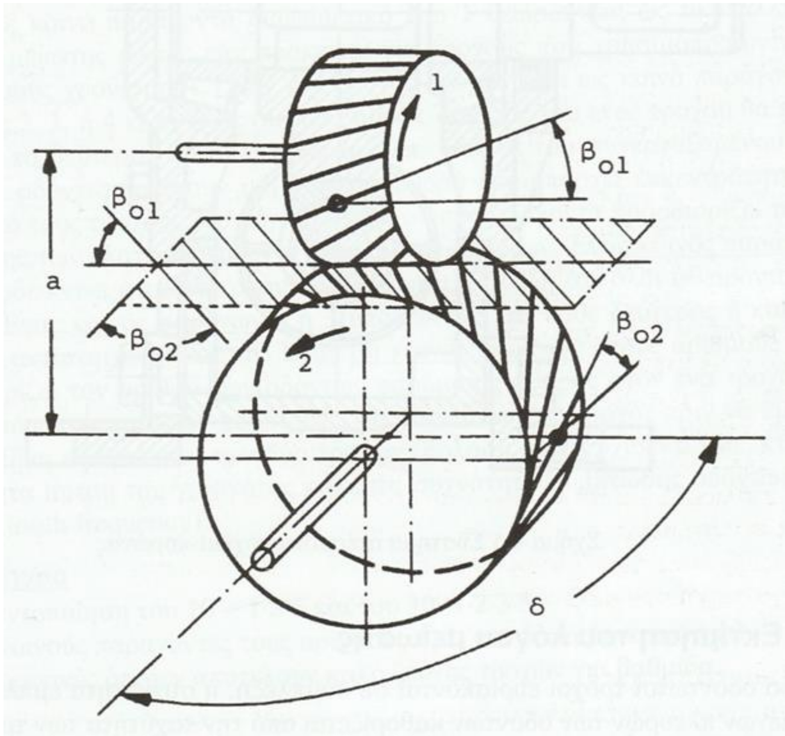
4.2.3 Helical wheels of incompatible axles

These gears are cylindrical and have incompatible axes at a distance a, and an angle d, which is such that: δ =$\beta_{01} + \beta_{02}$.

Where: $\beta_{01}\ and\ \beta_{02}$, are the inclinations of the wheels' teeth. In general it is β_{01} =β_{02}.

The teeth of the two cooperating helical wheels of compatible axes have a symmetrical contact, which turns into a linear one as the wear of the profiles increases. The wheels can carry small loads due to the way they are in contact, so they are used only for transmission and are not suitable for power transmission.

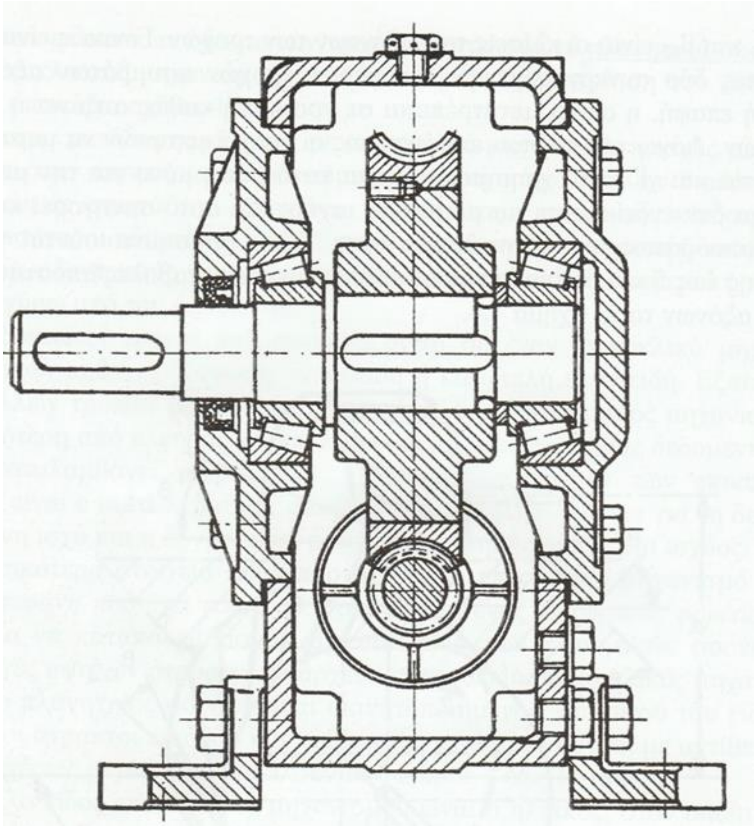
Finally, they are used for gear ratios up to 5 and are not sensitive to small changes in distance or inclination of their axles, sketch 4.4.



Sketch 4.4 Stage of helical wheels incompatible axis

4.2.4 Endless screw-crown system

This system, sketch 4.5, consists of an endless screw which is essentially a screwdriver of a suitable thread of one or more principles, which cooperates with a cogwheel in such a way that their cooperation resembles the way in which a screw involves with its nut. Their axis is usually perpendicular, although they may form another angle. This system is suitable for transporting large loads and for large gear reduction ratios up to 200:1.



Sketch 4.5 Endless bolt-crown system

4.3 REDUCTION RATIO ESTIMATE

When two gears are in coupling, the frequency of coupling of the working sides of the teeth is determined by the speed of the wheels and the number of teeth. The estimation of the most efficient and appropriate reduction ratio (gear ratio) of the wheels can be achieved by factorizing the number of teeth and determining the maximum common divisor. Apart from the timing mechanisms and some special applications, an ideal pair of wheels should not have numbers of teeth with a maximum common divisor other than a unit. When 1 is the only common factor between the numbers of the teeth of two cooperating wheels, then one tooth on one wheel must couple with all the teeth of the other wheel before it encounters the same tooth again, which will only happen if the tool numbers are prime to each other. This ensures the same wear on each tooth of both wheels.

When the wheel numbers have a common factor other than 1, then one tooth of the small wheel, in its multiple rotations, will couple with N-ost teeth of the other wheel (where N is the largest common factor). When this happens each N-ost tooth can wear out differently from the others. This means that the waveform of the wheel coupling will be different or it can be deformed and its transformation into a spectrum with FFT (Fast Fourier Transform) can produce strange wheel coupling frequencies.

The number of teeth in each wheel can be factorized to determine if the tooth numbers have a common factor other than 1. Wheels with a common factor other than 1 are considered unsuitable for a reduction step (unless they are wheels used in timing mechanisms). When a pair of wheels has the numbers 2, 3 or 4 etc. as a common factor, a specific tooth of one wheel will couple only with the second, third or fourth etc. teeth of the cooperating wheel. These teeth will have more wear if there is eccentricity on one of the wheels. The size of the joint determines the appearance of defective or worn teeth. A common factor equal to 1 indicates that each factor wears out, i.e., that all wear out evenly. A common factor 2 or 3 indicates that every second or third tooth respectively wears out. A non-common factor in tooth numbers determines the number of teeth of similar wear on one wheel and the number of revolutions the other wheel must make before the same teeth on both wheels couple again (i.e. before the new cycle begins). The frequency of this event is called the frequency of tooth coincidence (HTF = hunting tooth frequency).

Example

Factorization of $10 = 1 \cdot 2 \cdot 5$ and of $30 = 1 \cdot 2 \cdot 3 \cdot 5$

They have common factors the numbers 1, 2 and 5 and non-common number 3.

It is obvious that they consist a poor pair of wheels for a step.

4.4 WHEEL QUALITY CONTROL

Let's consider two wheels with numbers of teeth Z_1 and Z_2. The angle of the rotation does not correspond to the angle of rotation of the second $\varphi_1 = \varphi_2\ (Z_1/Z_2)$, as theoretically predicted, but at a value close to it. The difference between the predicted value for φ_2 and the actual value is called a transmission error.

The causes of the transmission error can be the construction errors of the wheel and the possible deformation of the tooth due to the load. Transmission error is the main source of stimulation of the vibrations caused, which are responsible for the dynamic loading of the gears. The larger the transmission error, the more intense the occurrence of noise and vibration which clearly adversely affects the strength of the mechanical parts and the performance of the transmission. This includes quality control that is the method of ensuring a maximum level error, according to the requirements of each construction.

Types of errors:

1. Construction errors such as: cutting tool profile errors, placement errors and unevenness in the step, indentations with rotation at a certain angle in relation to the predicted (wring angle), incorrect profile of thick teeth due to cam or incorrect cutter distance.
2. Simple errors such as: step (in the initial or basic cycle), tooth thickness, profile, eccentricity.
3. The superimposition of the above errors is the complex error

The composite error is measured by the rolling method.

It constitutes a method of measuring the complex error of a cogwheel by coupling it in operation with a standard precision wheel. In order to be a measurable phenomenon, one of the two wheels is allowed to move in the direction of the center of the wheels, under the influence of the repulsive force that occurs during the tight cooperation of the two gears.

Under these conditions, any abnormalities and generally any deviation from the theoretically predicted geometry of the wheel will have an effect on the magnitude of the force and will therefore cause corresponding disproportionate momentary displacement of the other wheel. The data is recorded with a PC program and then processed to classify the wheel in some quality category, according to the specifications, as in the system of classification of the accuracy of front cogwheels with specifications AGMA.

4.5 ELECTION OF THE APPROPRIATE SYSTEM

The satisfying operation of a cogwheel transmission system depends on the quality of its design and construction, the election of the appropriate type and size for the given application, the correct installation, and the proper use of the system as well as the completeness of the maintenance throughout of the useful life of the system. To select the appropriate type in a given application, we need to know the power transmitted through the system and the transmission relationship between the engine and the moving shaft. We must know:

- The power and speed with which the machine will move.
- The output speed as well as the engine power.

To calculate the power that each drive system must carry, we first check the degree of efficiency. In general, the efficiency of each stage of straight front gears, helical, double-helical teeth, of the conical with straight teeth, arches, subspecies, and conical wheels zero is very high above 0,90. The degree of efficiency of the endless screw-crown stage varies greatly since it ranges from 0, 20 to 0, 95 depending on the speed of movement, the angle of helix of the teeth, the factor of friction, the lubricant, etc. We consult the manufacturer of cogwheels in cases where we want to know the degree of efficiency of special drive systems with cogwheels.

The overall efficiency of the gear unit is derived from the efficiency of the gear the pairs use, when more than one pair is used. After the losses in the wheel steps have been calculated, the type of charge that will be received by the transmission system must be considered. For standardization and facilitation, the most common and frequently used machine tools are classified according to the type of work they perform in three types.

The first type concerns uniform-nonimpact load. The second type concerns the load of moderate impact, and the third type concerns the strong impact load. We define the load characteristics concerning the machine builder of an application from the circulating regulations. In this way we determine the Service factor Sf that is necessary for the calculations. Then we multiply the power (which is increased after it is divided by the efficiency of the transmission system) by the appropriate impact factor. The tables help us to choose these factors correctly depending on the type of engine, the load and the handling of the machine. Then, we divide the input shaft speed by the speed desired by the output shaft, to calculate the gear ratio of each drive system. Then we choose the transmission system that meets the speed limit and the highest requirements of the application in power or load.

After selecting the transmission power system, we check the transmitted power and compare it with the critical thermal power, i.e., the power above which there is a risk of overheating of the system and the use of refrigerants is required for their safe operation. It has been shown that wheels with a large module and small number of teeth are more likely to show surface cavities than to fail in the strength of their dangerous cross-section for the risk of failure.

On the contrary, sprockets with a small module and a large number of teeth are more risk of tooth breakage than the appearance of cavities. Therefore, to avoid cavities, a thicker lubricant should be used, a smaller module that ensures a lower foot and head height, a larger number of teeth and a lower speed. The use of better material and surface treatment is also required.

4.6 INSTALLATION AND OPERATION OF COGWHEELS.

While installing a cogwheel drive system, we should take care that it is well supported, precisely aligned and that the shafts or sprockets do not risk losing their alignment. We should always consult the manufacturer's installation and maintenance instructions.

High quality mechanical-elastic joints are necessary for the connection of the outlet shaft of the engine with the inlet shaft of the engine. Small angular or linear deviations of alignments can be addressed by the mechanical-elastic joints themselves. In some cases, additional torsional stresses at start-up or during momentary overloads can be compensated by using certain types of mechanical-elastic joints.

The correct and controlled loading and straining of the drive systems with cogwheels is an important factor for the smooth and safe operation for a long time, without any problems. For this reason, when a system is installed and put into operation, it is important not to submit it to unforeseen load conditions (extreme loads or long-term high load performance).

Torque adjustment clutches are available as an option in some manufacturers' transmission systems. Their use is particularly suitable in applications where there is strong risk of overloading the system or polarization of the machine, which inevitably leads to engine overload.

Gear systems are carefully designed to provide adequate heat dissipation under normal operating conditions. Special care is recommended so that the transmission systems do not operate with loads or in places where the temperature developed by the lubricants exceeds the prescribed. We must consult the manufacturers, in places where environmental conditions can limit heat dissipation from it.

Some manufacturers cover the inner parts of the sprocket systems for protection, with a layer of polar molecules to prevent rust. It is not necessary to clean these elements before the lubricant is entered, as they are usually molecules that dissolve the lubricants. Therefore, we simply fill in the required amount of suitable lubricant inside the gearbox of the transmission. We always check whether the manufacturer has added lubricant or not into the hull where the mechanism operates. Systems that include bearings, require the grease to be properly inserted and checked.

Systems containing pressure lubricant, when delivered and set into first operation, must be checked to determine if the lubricant is under pressure. When the system has a build-in pressure gauge, the indication pressure must be in accordance with that specified by the manufacturer. If the manufacturer does not specify an appropriate value, we should be aware that the pressure should be up to 1 to 2 bar with lubrication tank temperatures of about 70ºC. We also adjust the relief value to the pressure that we want the oil in the circuit not to exceed, if we want to keep it at a value specified in the user manual.

Each new system is put into short-term operation by the manufacturer himself, a process that is included in the inspection of the new construction. However, for laying the machine under the actual operating conditions, it is recommended that the system operate under part load for 1 or 2 days in order to form the final friction conditions of the gears. At the end of this period, the load may be gradually increased to its default value.

After the drive system has been operating at full load for 2 weeks, it must then be stopped to remove the lubricant and the lubrication tank to be cleaned. Because the original lubricant has not aged but simply contains chips, it is possible to be filtered well, checked and reused. When the hull of the gear system has been drained from the original lubricant, we fill it with mineral oil SAE10 without additives up to the indicated level. The system is put into operation, reaches its operating speed and stops abruptly. We drain the lubricant without additional materials and place the prescribed lubricant from the manufacturer to the indicated level.

After this first oil change, an oil change every 2500 hours or a period of normal operation of 6 months is recommended. This function may be abnormal when high ambient temperatures develop in combination with intermittently high loads, which result in the temperature developing in the hull of the mechanism being particularly high while the hull cools sharply. This process can cause water vapor to form on the inner surface of the hull, destroying the lubricant and creating a muddy material. Under these operating conditions, or during the continuous operation at temperatures constantly above 70ºC or even if the transmission is exposed to an extremely humid atmosphere, oil changes must be made every 1-2 months depending on the results of the regular inspection of the lubricant. Synthetic oils, especially hydrocarbons, can be used to increase the life of the lubricant. Lubricants intended for use in airtight gearboxes or driving systems with cogwheels are required to be of high quality, very clean, non-corrosive to gear wheels or bearings, inert and non-reactive, non-reactive foaming and not to contain sandy materials or materials that create friction, etc.

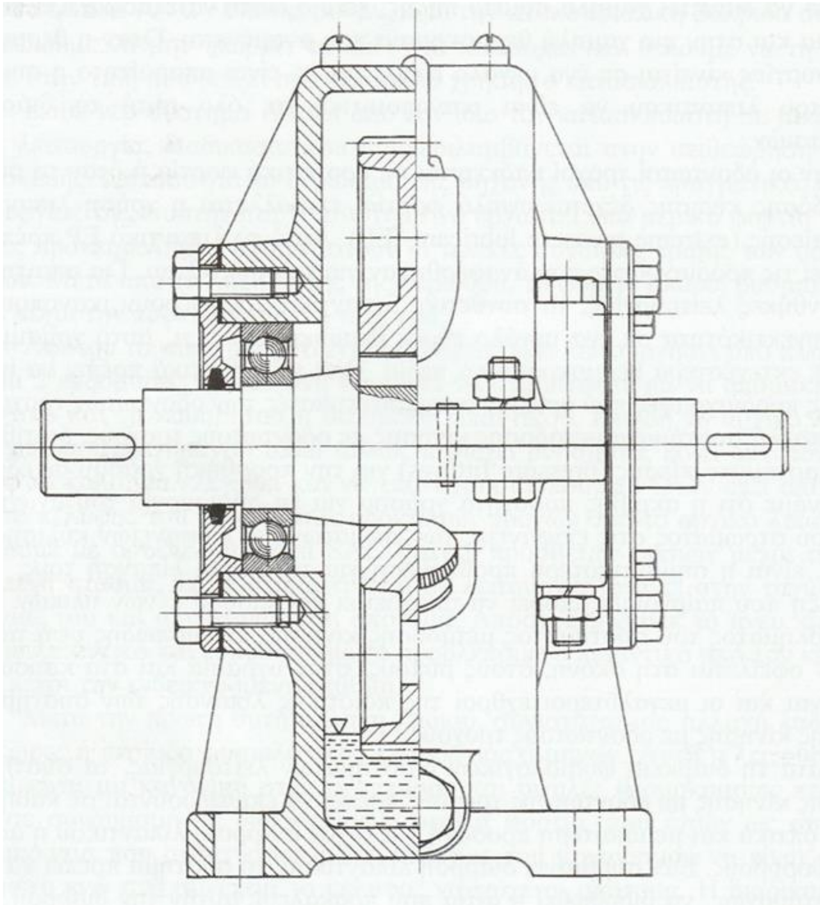
High resistance of the lubricant to oxidation is necessary for high operating temperatures. For low temperatures, the lubricant must have a low freezing point, such as to ensure its flow even at the lowest temperatures expected. It is essential that the consistency of the lubricant is satisfying throughout the temperature range, when the operating temperature is in a wide range of values.

When the gears are subjected to impact loads or when the drive system receives high loads, the use of high-pressure lubricant is required (extreme pressure lubricant-EP). This lubricant EP must meet the specified specifications for mineral oils. Synthetic lubricants offer satisfactorily high consistency over a wide range of temperature, for more demanding operations, which is why they are used in wider temperature ranges. All lubricants must meet the specifications set by the gear manufacturers.

In many driving systems with cogwheels, special pressure fittings are available to add grease to bearings. Do not forget that the exact amount of grease for the creation of the necessary lubricant layer on the surfaces of the bodies and rolling surfaces of the bearings, is the most important condition for their proper lubrication. Every precaution we take can prevent foreign materials from entering the transmission system housing. The muddy texture of the lubricants is due to the dust, the pollutants, the humidity and the exhaust gases. These are also the biggest enemies of quality lubrication of cogwheels.

During normal operating intervals, cogwheels must be inspected on a daily basis visually and with special care for leakage of lubricants or unusual noises. If a lubricant leaks, the system must be switched off to correct the cause of the lubricant leak and finally to check if the level is correct.

If various unusual noises occur, the system must be switched off again until the cause of noise is identified and rectified. We check all lubricant levels at least once a week. The temperature of the transmission is that of the lubricant working inside its hull. Under normal conditions, the maximum operating temperature does not exceed 85ºC. Transmission systems that receive lubricant under pressure, carry a working lubricant filter which should be regularly cleaned. If it is deemed necessary to shut down the system for a period or more than a week, it must be put into operation without load, for at least 10 minutes each week. This short-term operation keeps the gears and bearings in the lubricant and helps to prevent oxidation of metal surfaces due to moisture owing to temperature changes.



Sk. 4.6 Lubricant level in a gearbox.

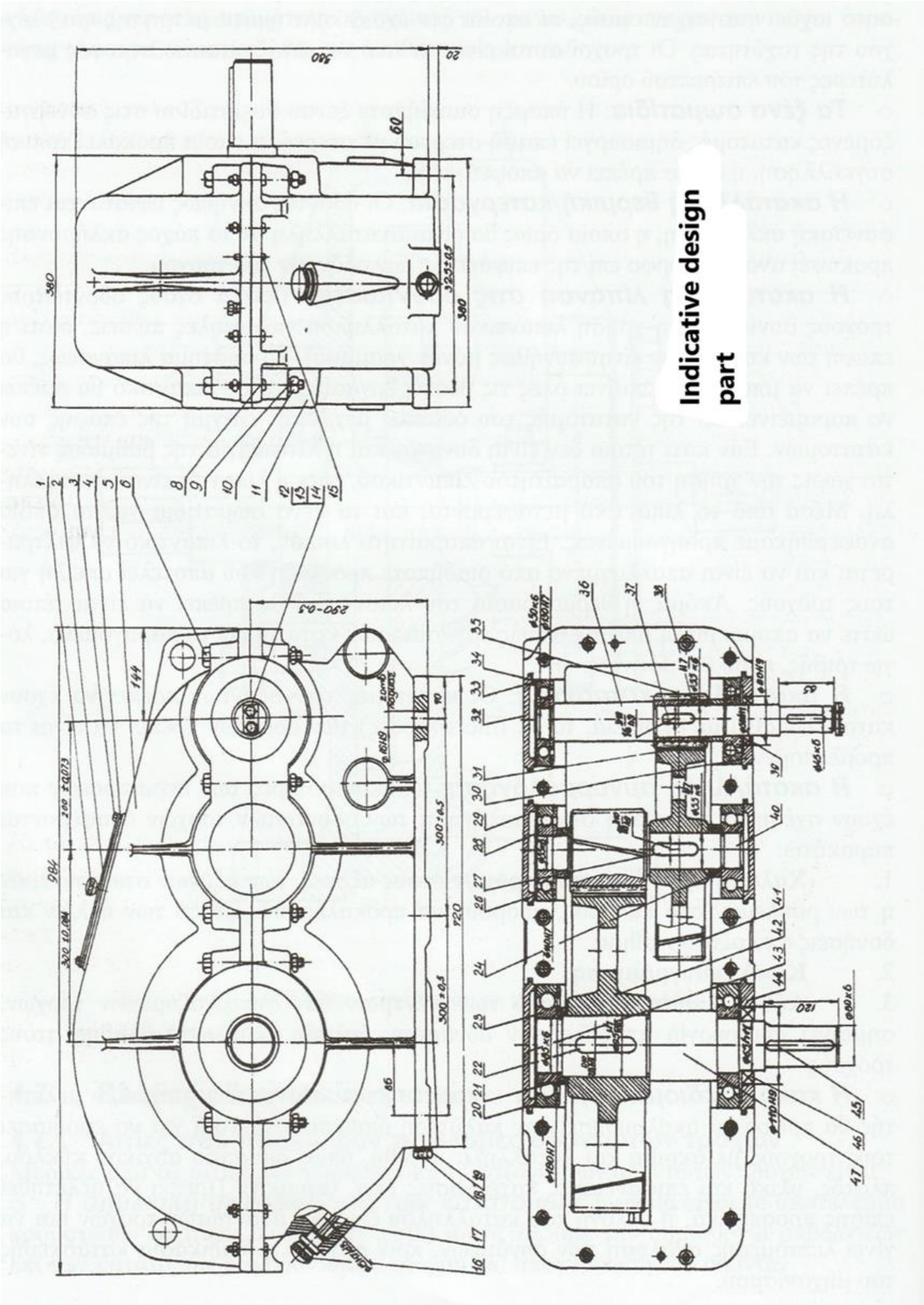
4.7 DAMAGES OF COGWHEELS

4.7.1 Reasons that cause the damages of cogwheels

Some of the reasons that cause the cogwheels’ failure are:

- The point overload of the profiles: which is the result of impact loads. As a result of this we have the creation of steaks on the profiles of the cogwheel and the stripping of the tooth.
- Excessive speed of rotation (beyond the allowable): this cause applies to mechanisms that do not have a system for measuring and controlling speed. These wheels are likely to operate at speeds greater than the permissible limit.
- Foreign particles: the existence of any foreign particle in the cooperating profiles creates solid-to-solid contact, which causes local bonding which should be avoided.

- The inappropriate heat treatment: the teeth are usually subjected to surface hardening, which however will be inappropriate if the hardening thickness is uneven on the surface of the wheel teeth.
- Improper lubrication in serrations: generally, in cogwheels the use of a lubricant suitable for high pressures is recommended, because the contact of the profiles is only linear. The lubrication system should be able to lubricate all work positions and the lubricant should remain on the tooth profile until the contact of the profiles. If such a thing is not possible and the operation of the stage is done without the use of the necessary lubricant, then the lubrication is inappropriate. Through the lubricant the foreign particles which we mentioned earlier are also transferred. It is therefore essential the lubricant to be filtered and free of any impurities that are a threat to the wheels. Still, the temperature of the lubricant should be such as to remove the produced heat from the working positions of the profiles, due to friction.
- Inappropriate device: the tooth profiles must be precisely made, and the tooth fillets must be as intended.
- Improper assembly: most of the cases related to the incorrect assembly of the cogwheels are mentioned below:
  1. Loose attachment, of the wheels to the shafts, the shafts to the bearings or the bearings to the hull, can cause, among other things, vibrations from imbalances.
  2. Poor alignment.
  3. Wrong distance between the centers of the cooperating wheels means the creation of unpredictable forces, i.e., the occurrence of damage to the wheels.
- The poor design: in the construction of cogwheels, the designer should follow the best available technique to supply the wheels with accurate and appropriate sizes, such as initial cycle diameter, width, material, and surface treatments (e.g., heat). The selection of the appropriate pair of gears must also be carefully considered and a detailed analysis of the forces should be made before the construction process of the mechanism begins.



4.7.2 Criteria for malfunction of the cogwheels

It is easy to understand that the good and efficient operation of a cogwheel depends on whether it operates according to (or with small tolerances) or not according to the manufacturer's specifications.

- Power loss: the frictional power loss and its conversion to heat is a good indication of how wheels work. This amount of heat is measured and compared to the pre-calculated. The measured heat must be less than or equal to the pre-calculated to say that the mechanism works efficiently.
- The oscillations: a gearbox must produce oscillations within the prescribed limits. Dynamic balancing is one of the first steps that will be taken for the rotating parts of the gearbox. Oscillation problems can also arise from the following:

1. A tire that sticks to an unbalanced position.
2. Structural errors or operational errors of the profiles.
3. Improper assembly or alignment of the box components.

Of course, the good or bad operation of a machine component can be determined with the help of monitoring-measuring systems, after repeated measurements and observations. Such measurements are those referring to the machine maintenance,

- Measurement of shock pulses.
- Detection of the vibration-shock level.
- Fault detection with thermography.
- Study of the properties of the lubricant.
- Measurement of the parameters of the production process.
- Visual inspection.
- Other non-destructive structural techniques.

According to AGMA (American Gear Manufacturers Association) a properly designed and precisely constructed pair of cooperating gears, which has been applied, installed, and operated properly, requires an initial period of smoothing of the surfaces of the cooperating tooth profiles. Then, if the gears are properly lubricated and not overloaded, the combination of rolling and sliding of their teeth causes abrasion of their surfaces and gives them a surface protective layer in the form of a sanding surface. Provided that they will henceforth operate under normal operating conditions, the wheel teeth will suffer very little or zero wear. Despite the correctness of the above sentence, there is a possibility of failure of the teeth due to high wear of the cooperating surfaces or due to breakage of the teeth. In several similar cases, recognizing the possible occurrence of a problem has helped to treat and cure the evolving wear before it causes some extensive damage.

4.7.3 Fatigue and aging of the wheels at surface pressure

Fatigue is mechanical corrosion resulting from contact forces due to load. It is the fatigue of the surface and causes failure of the material resulting from the repeated stresses of the surfaces or the stresses inside the tooth, below its surfaces which exceed its permanent strength (Pitting process). It is characterized by removal of the metal and the formation of cavities. Cavities can be small in size and maintain it, but also increase it gradually. In this case the constant expansion of fatigue causes joints of adjacent cavities and increases their size. There is also a case that cavities from the beginning are large in size.

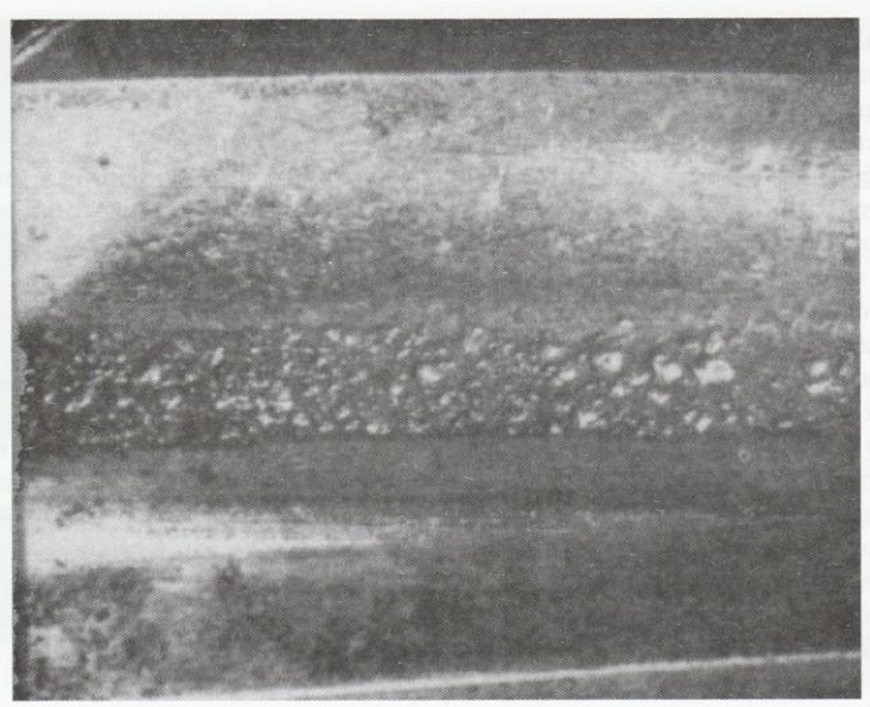
The Hertz theory is applied for the first test of cogwheels in surface pressure resistance. Simple static theory argues that aging will be worse where the stresses are highest. Mechanical corrosion initially occurs very close but not exactly up to diameter of the initial cycle, where the slip velocities are low or zero. It is no coincidence that sometimes especially on wheels that have worked for many hours, the area near the diameter of the original cycle is blackened. This of course warns us that the wheel needs immediate attention.

Fatigue depends on the tendency exerted on the teeth and is therefore a relatively slow process, which in most cases does not develop further.

There are the following types of cavities:

The initial cavities. These usually occur in places with high stress concentrations and are due to local physiological wear of the surfaces (local polishing). It is the type of fatigue that can occur at the start of the operation of the cogwheel and extends only until the large overloaded areas of the tooth surface are reduced, resulting in the lifting of the load from a sufficiently large surface without further downgrading of its quality. Usually appears in a narrow band below the original wheel cycle. This form of fatigue is not dangerous, as it can be repaired. For maintenance: high tension positions must be removed or smoothed by sanding and polishing the teeth subject to friction.

The destructive cavities. The cavities created here are much larger than the previous ones and always cause destruction of the profile. They are a continuation of the initial cavities but gradually increase in size and number until the smooth operation of the pair of cogwheels is altered. It is obvious that there is no hope of restoring the profile. For maintenance: in the early stages destructive cavities



Sketch 4.7 Appearance of cavities on the helical cogwheel surface.

can be detected after sanding and polishing the surface of the teeth. If polishing fails to slow down the rate of surface destruction, one solution is metallurgical hardening of the surface, which often stops the spread of wear. In some cases, the use of high-pressure lubricants has been successful in stopping the spread of this fatigue.

The local cavities. They create a local fracture of the profile in the form of sculpting and is essentially the same wear as the previous one, except that the inclusions created on the surface of the tooth are much larger but locally limited. It results from superficial or internal defects in the tooth material or from extreme internal stresses due to the thermal processes of the components. The cavities that are formed are deeper and cleaner than those of common fatigue. For maintenance: in the early stages the use of a very high-pressure lubricant can slow down the rate of development of the phenomenon.

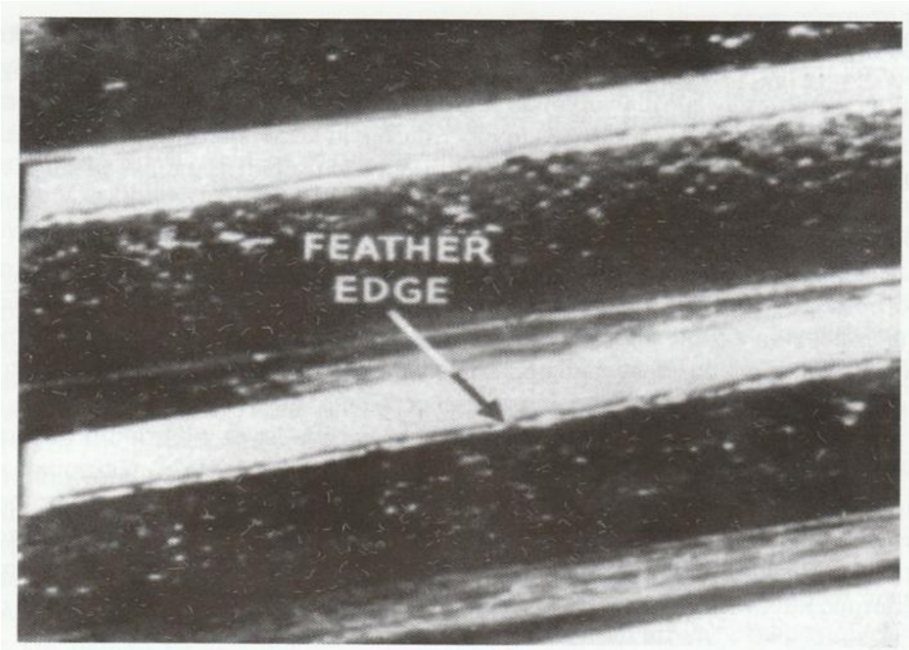
The micro cavities. They cause micro-corrosion and micro-fatigue followed by fatigue and aging. It has similarities to conventional fatigue but occurs at slightly lower loads. Unlike conventional fatigue, it tends to spread and progress. It can start anywhere on the tooth surface. The onset is due to contacts of large stresses, which through frictional forces create locally high pressures between the cooperating teeth, which enhances the process. The basic condition for the start of micro-corrosion is that the size of the surface irregularities (roughness) of the tooth should be of the same or larger size usually than the thickness of the lubricant layer, which is usually of the order of 1 μm. Lubricants at high temperatures tend to reduce the thickness of the lubricant layer and thus increase the likelihood of micro-corrosion.

The main reasons for the appearance of cavities are poor alignment, local overloading of cogwheels and high speeds. In conclusion, due to the local pressures with the presence of lubricant, the cavities appear in the area of the rolling cycle (where the sliding speed is very low) of the wheels, which are favored by the thin lubricants that wet deeper the roughness and prevent the existence of air, which otherwise acts as an inhibitory element. Due to the fatigue of the material from the high voltages, cracks are created on the surface of the tooth where the lubricant enters under high pressure, which is due to the impenetrability that is created locally in the lubrication area and the minimum compressibility of the lubricants. Over time these cracks widen.

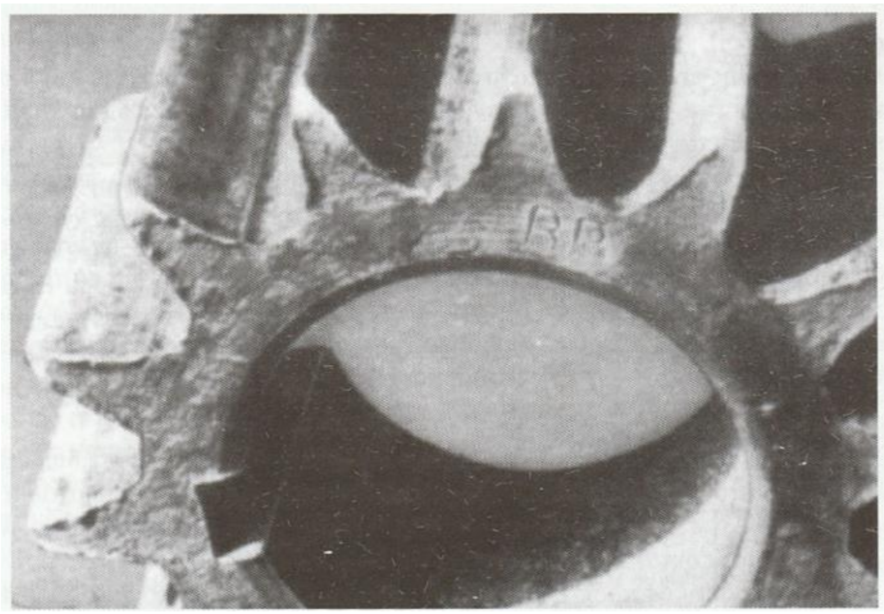
4.7.4 Breakage of teeth by bending load

Fatigue breakage is the most common type of failure and results from repetitive flexural stresses that exceed the strength limits of the material. These stresses are born through defective design, overload, misalignment, or unexpected increase in stresses in the notches or surface or imperfections under the surface of the material. The fatigue breakage comes from a crack in the material. Cracks in the teeth would not be a problem if they did not expand. Cracks are usually formed at the base (fillet) of the gear. The crack starts from an increased surface tension in a dangerous cross section of the foot, on the side of its tensile (working side) and develops rapidly causing total failure along the foot of the tooth or in the diagonal direction towards the head of the tooth with effect or detachment of the entire tooth portion. In helical wheels, detachment of the whole tooth is not common, but it is possible to have detachment up to 1/3 of its width.

The breakage due to intense wear. The crack of the tooth or its breakage is the final result of the continuous deterioration of the quality of the surfaces of the gear wheels. Catastrophic cavities can remove large amounts of metal from the tooth to reduce its strength below the safe operating limit and lead to breakage.



Sketch 4.8 Cusp development to the head circle that reveals plastic deformation and flow of the material towards the head of the tooth.



Sketch 4.9 Overloaded teeth from soft material that reveal plastic deformation and flow of the material towards the tooth edges.

Breakage from overload. It is a relatively common type of failure resulting from sudden overload. It has been observed that in some gear designs we make a basic assumption that anywhere there are specific parallel power transmissions the load is evenly distributed between the different paths. If this assumption is not correct, power unequal distributions and load increase capable of causing damage can be observed.

Lack of alignment that causes stress to collect on one side of the tooth is most often the main cause of breakage, but overload breakage can also be due to welding of teeth due to bearing failure, bent spindles, or large sections of foreign material that enter the area of tooth coupling.

An extreme case of load unevenness can occur in the assembly of hydraulic lifting mechanisms, where there can be many electric motors which will work in parallel through gear units, with the aim of raising or lowering loads of thousands of tons as happens through rotating pinion which cooperate with vertical gear rules.

4.7.5 Tooth surface wear and maintenance instructions

The term wear refers to all types of material losses on the working side of the tooth.

<u>Friction wear</u>

Friction wear is the normal wear due to slipping of surfaces on moving parts of machines. During the normal process of wear the final outer layer develops on the surface each time. As long as this layer remains stable, the surface wears out at a normal rate. If the layer is removed faster than it is produced, the rate of wear increases and the maximum particle size increases accordingly. The presence of certain substances in the lubricant, can increase the friction wear by more than one order of magnitude. Although complete failure of the material seems unlikely, machines of this type wear out and are destroyed relatively quickly resulting in a rapid increase in wear particles.

<u>Wear from material removal</u>

This type of wear is due to the formation of small particles in the lubricant when one surface penetrates another. These particles are produced due to poor alignment or breaking of hard surfaces, the edges that form cause sections of any softer material to be cut. A different way of forming them is when rough particles are attached to a soft surface and then penetrate it by cooperating with another harder one. If the dimensions of these particles are of the order of a few μm in length and one μm in width, then the only problem that can occur is in the pollution of the lubricant. The increase in the number of particles and their length is a serious indication of impending failure of the lubricated engine components. The particles formed by this technique of penetrating the relatively hard surface into the softer, are very similar to the chips (grits) when removing material on the lathe. The size of the particles formed varies from 2 to 5 μm in width and from 25 to 100 μm in length.

Wear due to rolling

Wear due to rolling fatigue can be caused by three different types of particles: surface fatigue fragments, spherical particles, and small plate particles. The released particles leave cavities on the surfaces. The particles have a maximum dimension of 100 μm during their initial formation process. The laminated particles have a maximum dimension with a thickness greater than 10: 1. Fragments are caused by cutting a portion of the worn surface of a bearing and creating a cavity. An increase in the number of fragments or their size is a sign of wear. The wear of rolling of one surface over another does not always give rise to spherical particles and can have a variety of origins. The appearance of the particles is important because they are perceived first when any serious damage occurs inside the lubricated parts of the engine. The particles that are in the form of a laminate are particularly thin and can be formed during the passage of a typical wear particle, which rolls on a surface. The laminate-shaped particles may form during the life of a wheel or bearing, but after the onset of fatigue breakages, their number increases sharply.

Combination of wear by rolling

This combination occurs upon contact of cooperating gear surfaces. These larger particles result from tensile stresses on the surfaces of the gears, acting in such a way that the debris due to fatigue, penetrates deeper into the teeth of the wheels causing deep cavities (pitting phenomenon). The breaks made in the cogwheels do not produce spherical particles. Crankshaft wear is caused when operating at high speeds or carrying heavy loads. The maximum amount of heat generated destroys the necessary lubricating membrane of the cooperating wheels and causes weld of the teeth of the cooperating wheels. The surface becomes rougher and rougher and the rate of evolution of the wear phenomenon increases. From the moment this type of wear begins in driving systems with gear wheels, the evolution is given and concerns all teeth. There is a great variety in the characteristics of the particles formed in terms of wear velocities due to simultaneous rolling and sliding of two surfaces. These particles bear a strong resemblance to those produced by wear inside the roller bearings. The larger dimension of the particles to the smaller, varies from 4:1 to 10:1. The larger particles are produced by tensile stresses on the surface of the gears, causing surface cracks that gradually extend to greater depths of the teeth. The sizes of the particles formed from 20 μm to 2 μm are obvious.

Wear due to strong slip

Excessively high loads of force or heat cause this type of wear on a cogwheel system (e.g., gear unit). Under these conditions, large particles separate from the worn surfaces, causing an increase in the wear rate. If the stresses applied to the surface increase more, we reach a second phase of the wear effect. The material on its surface fails consistently the appearance of cracks and the expansion of wear on the material until its complete destruction. The particles created by the strong slip cause damage to the surface and have dimensions greater than 20mm. Some of these particles have grooves on their surface due to slipping. They usually have straight edges and the maximum to minimum ratio is 10: 1. The presence of many contaminants from the metal components found in lubricants can be attributed to more than one root causes. Unfortunately, with this method we cannot find out exactly where each polluting component of the lubricant came from or at what point the components have deteriorated. For example, a sample of lubricant may contain copper, which can be interpreted as possible wear of a roller bearing. The analysis cannot confirm this hypothesis nor identify which of all the bearings is the one that showed the fault. Due to all these limitations, the tribology has little use as a tool for predictive maintenance. An exception to this is its use as part of a lubrication improvement program aimed at increasing the life of lubricants (oils and greases) used in engines and transmission systems. Particle wear analysis is an excellent tool for analyzing the suitability of lubricants and can be used when trying to identify the true cause of premature lubricant wear.

Normal wear is the slow loss of metal from the cooperating surfaces to the extent that it does not affect the satisfactory operation of the gears during their intended life. For maintenance: most manufacturers of pre-assembled drive units offer frequent cleaning of the housing in which the gear mechanism is located, in order to remove all metal particles and eliminate the possibility of their movement between the wheels.

Abrasion damage of the surface is the superficial injury of the teeth due to fine particles which pass through their points of coupling and cooperation. For maintenance: whenever abrasion damage is detected, the system must be switched off immediately. The lubricant needs to be disposed of.

Surface carving is a serious form of surface abrasion wear, characterized by fine lines on the cooperating surfaces in the direction of slip. For maintenance: we always make sure that the cogwheel, gear, and diodes of the lubricant are completely free from the presence of foreign particles.

Surface wear from overload. It is a form of wear observed in high load and low speed conditions, on both hardened and non-hardened gears. The medal appears to be progressively removed in thin layers or hulls, leaving the surfaces as if sharpened. For maintenance: the only treatment for surface wear overload is to reduce its rate of progression, by reducing the load.

Leak in the plastic area. The leakage of the material in the plastic area is the degradation of the condition of the surface, under the exercise of large loads with the creation of edges. It is mainly caused in soft metals but can also occur in fully hardened or superficially hardened metals. It is associated with overloads or improper lubrication and usually leads to complete failure unless the material has high resistance to harsh operating conditions. The formation of edges is a special form of leakage in the plastic region that occurs on the surfaces of the teeth of a superficially hardened subspecies pinion or of a brass endless screw. For maintenance: the gears must be adjusted so that the load is evenly distributed on the surface of each tooth. The use of high-pressure lubricants can reduce the rate of surface quality degradation.

Engraving

Involves breaking the lubricating layer resulting in metal-to-metal contact during dental cooperation. Creates violent fusion and then separation and plastic flow on the two cooperating surfaces. Engraving is the rapid peeling of tooth surfaces due to the cutting of small particles, which are glued together as a result of metal-to-metal cooperation and leave a surface characterized by cracks or creeps and grooves, with all the marks on its sliding direction. May also be related to the possibly very small thickness of the lubricating layer or to the imposition of higher than permissible stresses or to high sliding velocities between the cooperating surfaces. On some wheels that spin at low speeds and at high loads, the use of a good lubricant can restore previously worn surfaces from cold etching (in the absence of a sufficient lubrication layer).

It usually starts at the surface where high surface pressure and high-speed slip on or very close to the tooth head are combined. For maintenance: the repair of the light engraving is often achieved by the use of a high-pressure lubricant or it may be necessary to smooth the tooth surfaces or to harden the surface of the wheel teeth.

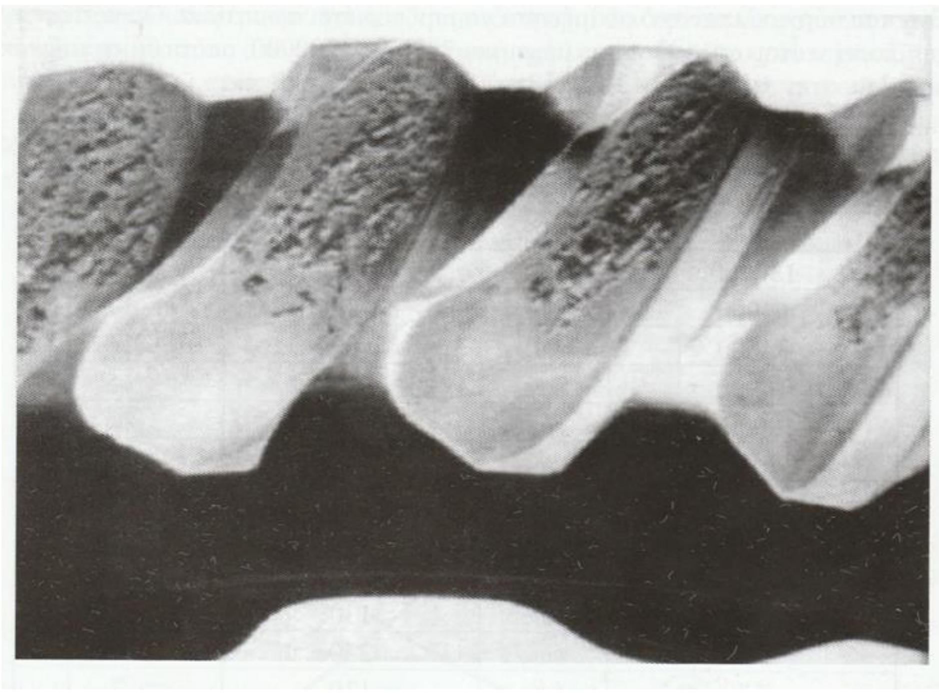
Corrosion wear is the wear of the surface associated with the action of chemical elements, such as acids, moisture or contamination of lubricants. It can occur under a variety of circumstances. If the lubricant is contaminated with acids from the environment, the teeth of the wheel will begin to show small cavities. The oxidation of the surfaces due to the entry of water into the lubricant due to condensation of water vapor, the excessive humidity as well as other factors cause similar consequences. For maintenance: remove the source of wear that is the contaminated lubricant and wash to clean. We make sure that the new lubricant is completely clean.

Grip from overheating

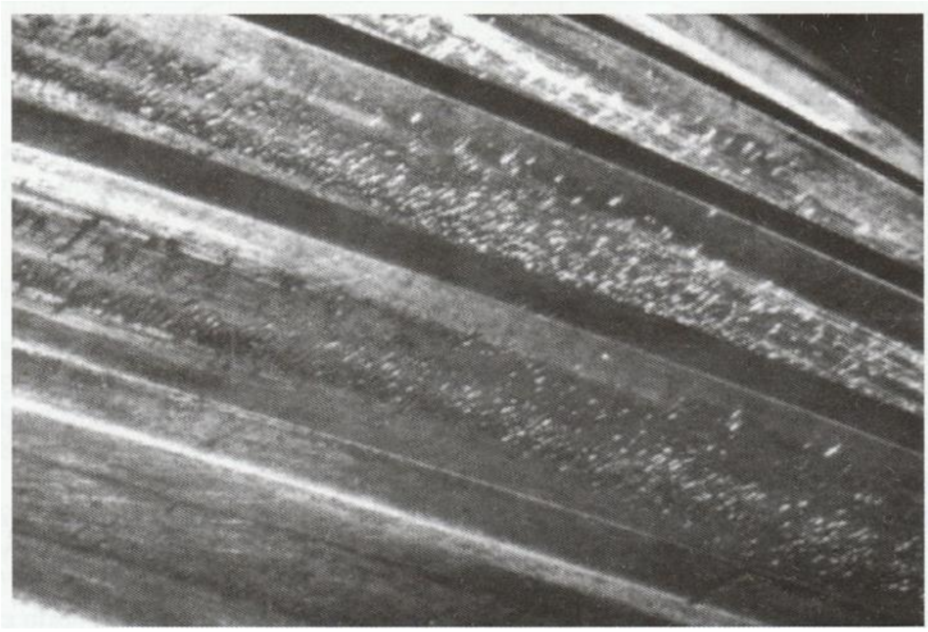
Cooling of gearboxes is rarely a problem for small sizes because the surface area in relation to the power is large. When it comes to large sizes, there is usually an external (additional) cooling system to control the oil temperature. Overheating can occur when based on natural heat transfer the heat produced is greater than predicted. It is characterized by temperature discoloration of the contact or adjacent surfaces of the teeth and is the result of an extreme increase in temperature either from external sources or from high friction due to overload, speed higher than expected or unsatisfactory lubrication. For maintenance: we first examine the lubricant, and we must be sure that the gears will not rotate outside the specified speed and load range.

Foreign particle detection

Detection of foreign particles is one of the oldest techniques we use to get an indication of damage. Magnetic plugs (attracting iron filings) were of limited use because they were used only when the lubricant change was scheduled. Modern particle counting techniques are very effective in giving accurate quantitative calculations of the condition of the lubricant and should be used if the gearbox bearings carry heavy loads and are therefore very vulnerable to contaminants or foreign particles present in the lubricant.



Sketch 4.10 Worm gear that presents stress corrosion.



Sketch 4.11 Tough foreign particles (pollutants) in the lubricant that cause the creation of prints on the teeth surface.

A new lubricant is expected to contain no particles. But in practice we can detect up to 7000 to 200000, so we would classify it in category 13 to 18, based on the following table:

Foreign particles in 100 ml of lubricant			category
500000	Up to	1000000	20
250000	Up to	500000	19
130000	Up to	250000	18
64000	Up to	130000	17
32000	Up to	64000	16
16000	Up to	32000	15
8000	Up to	16000	14
4000	Up to	8000	13
2000	Up to	4000	12
1000	Up to	2000	11
500	Up to	1000	10
250	Up to	130	9
130	Up to	250	8
64	Up to	130	7
32	Up to	64	6

Table 4.1 Classification of lubricants based on the number of foreign particles contained.

Serrated failure areas. The diagram below shows the different areas of torque, depending on the circumferential speed of the initial rolling point, where the sprockets fail or not.

Area 1: the wheel does not have enough speed to maintain the lubricant layer up to the contact position of the profile.

Area 2: is the safe operating area. The speed is enough to have a layer of lubricant in the working positions of the profiles. The life of the wheels is unlimited, as long as the lubricant is free of particles and chemical reagents. Sprockets must be designed to work in this area.

Area 3: here occurs rapid surface wear due to a combination of high loads with high speeds, which destroy the lubricant layer at the contact points.

Two other areas are mentioned here, such as area 4 where the loads are too high, and the teeth break and area 5 where fatigue (pitting) profile occurs.

Finally, we must mention that the cogwheels must not be charged with their full load from the first moment of their operation. The cogwheel technique requires their gradual loading up to full load and up to their maximum operating speed. Of course, this process is time consuming, but this delay allows the cooperating profiles to polish each other by removing foreign particles through their normal and within the predetermined wear limits of AGMA.

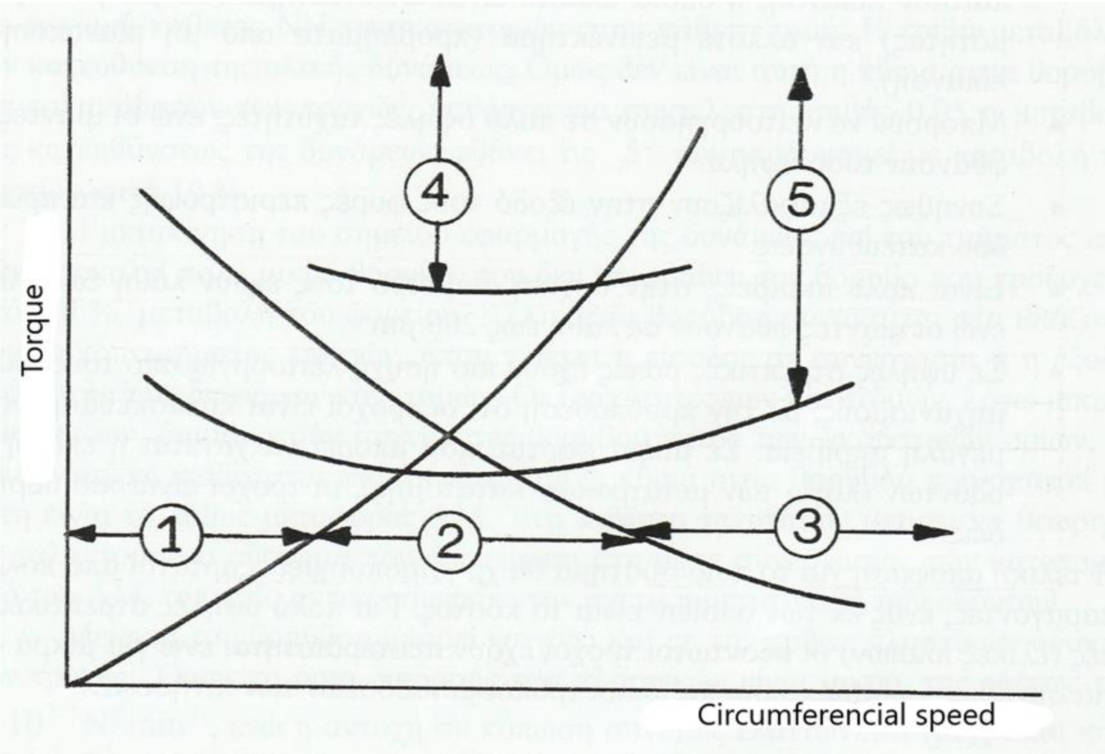


Chart 4.1 Areas of cogwheels' failure

4.8 CHARACTERISTICS OF GEARS WITH INDENTATIONS

Transmission and power transfer is not only done with gearboxes but also with other mechanisms. Belts (flat, trapezoidal, toothed), chains and various hydraulic, pneumatic, or electric machines can be used.

Because it is sometimes difficult to choose the most suitable transmission and power transmission system, and to help the designer in his work, they refer to the mechanisms made with cogwheels as follows:

- They are compact and very suitable when the engine-machine distance is short. For long distances the belts and the hydraulic systems are better.
- They are modern machines in the sense that with the constant gear ratio they ensure a constant number of revolutions required by many machine manufacturers. However, belt slippers and hydraulic systems have the ability to smooth out impact loads.
- At high torsional loads they are cheaper than electric or hydraulic mechanisms.

- Their axes remain aligned even at high loads. They require lubrication which is sometimes an advantage (heat removal) and sometimes a disadvantage (problem of unsatisfactory lubrication).

- They can operate at very high speeds while the belts do not reach that high.

- They usually provide rotation times in both directions at the output.

- They are very accurate; in their initial diameter they have errors up to 20 mm while the straps reach errors up to 200 mm

- At very high torque they have quieter operation than other mechanisms, provided the wheels are made with great precision. At small loads, where tooth contact may be lost (due to profile converters), the wheels are noisier.

The final decision on which system to use depends on many factors one of which is the cost. For very high torques (ship propellers) cogwheels take precedence, while for small loads other modes of transmission are usually used.

The noise in the gearboxes

Noise and oscillation problems in the gears are associated with the degree of dissimilarity of both the movement and the transmission of forces to the wheels, as the change in speed and force are the main causes of noise. The degree of dissimilarity is called the transport error (TE) and is defined as above, as the difference between the position of the machine's shaft if the gearbox was perfect (without construction errors or deformations) and the actual position of this shaft. TE can be measured either as an angular displacement or as a linear displacement along the energy line (in the basic or initial cycle). Angular deformation is usually measured, which is converted to TE in length of energy (when profile conversion is studied) or refers to the initial cycle (when the elongation angle of the tooth is examined). TE is directly related to the noise produced because it is the main cause of oscillations.

Noise or oscillation is created in a machine when one or more of the lower characteristics of the active force, the meter (size), the point of application, or the direction, changes as a function of time.

In the indentations various structural errors on the cooperating profiles are the cause of oscillations during the direct action (in the indentations evolving during the contact trajectory). This oscillation is transmitted through the wheels, shafts, and bearings to the gearbox housing, which on the one hand vibrates and creates noise, on the other hand it does not transfer oscillations to the rest of the engine.

In advanced gear, when the friction effect is neglected, the total force on the tooth is constantly exerted in a constant direction, which is the common perpendicular NN of the profiles in the vertical section. The friction changes the direction of the total force. However, this is not the main cause of noise and oscillations of the wheels as for a coefficient of friction 0.05 the change in the direction of the force reaches 5 degrees, which is equivalent to a change in the load by 10%.

The movement of the force application point on the contact section causes very little noise that does not exceed the noise resulting from a 10% change in load. Another cause of noise is found in the specific points of the contact department, where there is an entry into a collaboration or an exit from that of the various profiles. The roughness of the profiles due to their small dimensions create noise frequencies beyond the acoustic limits. From the above it becomes clear that the only main cause of noise that requires study is the transport error TE. A gearbox can be thought of as an oscillating system that is excited at the cooperating position of the profile under the TE, which represents the relative displacement of the teeth.

Noise reduction can also be done with the use of plastic cogwheels. However, the leakage limit of plastics is small of the order of 7•107 N/mm2, while the fatigue resistance is constantly decreasing (up to 90% of the original) as the number of charge cycles increases. Still the big advantage of plastic wheels is due to the very small elastic limit (3•109 N/mm2), which is about 1.5% of the corresponding steel. This allows the contact surface to be increased, which significantly reduces the developing surface pressure. This means that the profile is not endangered by Hertz pressures and that the control should only be done in bending in the dangerous section. In 107 load cycles the plastic gears have about 10% of the strength of the steel wheels of equal size.

The frictional heat generated is of great importance for plastic wheels. Lubrication is not essential for the operation of plastic wheels, but it does improve the transport of loads and remove frictional heat. The expansion of the plastic is ten times greater than that of the steel, and therefore the joys in the teeth must be large (in addition to these there is an increase in the volume of the plastic up to 1% due to humidity). Someone must consult plastic manufacturers when it comes to making plastic cogwheels.

Financial data

The cost of sprocket gears depends on many factors which must ensure satisfactory operation at the lowest possible cost. The satisfactory function consists of different things, in each application. For example, a simple and cheap gearbox is mentioned, which will have satisfactory operation when for a long time it does not show problems. In continuous production industries the cost of shutdown to repair a gearbox is much higher than the price of the gearbox itself. In this case the satisfactory operation is equivalent to a high degree of confidence for a very long time which of course are ensured by gearboxes with high initial cost.

The cost of a gearbox consists of:

1. The cost of materials
2. The cost of construction and
3. The cost of trade materials (bearings, sealants, lubrication system components, etc.)

In order to find the cost of the optimal gearbox for a new job and to evaluate the effect of the chosen design of the new gearbox on the rest of the construction, the gearbox must be designed, and a cost analysis must be done for all possible solutions from which the gearbox will result. Usually, companies get complete systems from gearbox manufacturers or build their own gearbox if of course they have the necessary tools-machines and the relevant experience. In this case the cost depends mainly on the materials and the quality of construction.

Material cost

Steels can be purchased in the form of rods (up to mm in diameter) or forged pieces (diameter over mm). The stronger the material (hence the more expensive) the smaller the dimensions of the wheels (i.e., they reduce waste). Many times, the question of the gearbox housing is asked, whether it will be cast or welded. For a small number of pieces, the welded is cheap. For large numbers of boxes, the casting ensures a cheap product as in fact the cast product may have approximately its final shape and require minimal machining. Cast boxes can be cast iron or steel and the weight reduction required can be made of aluminum.

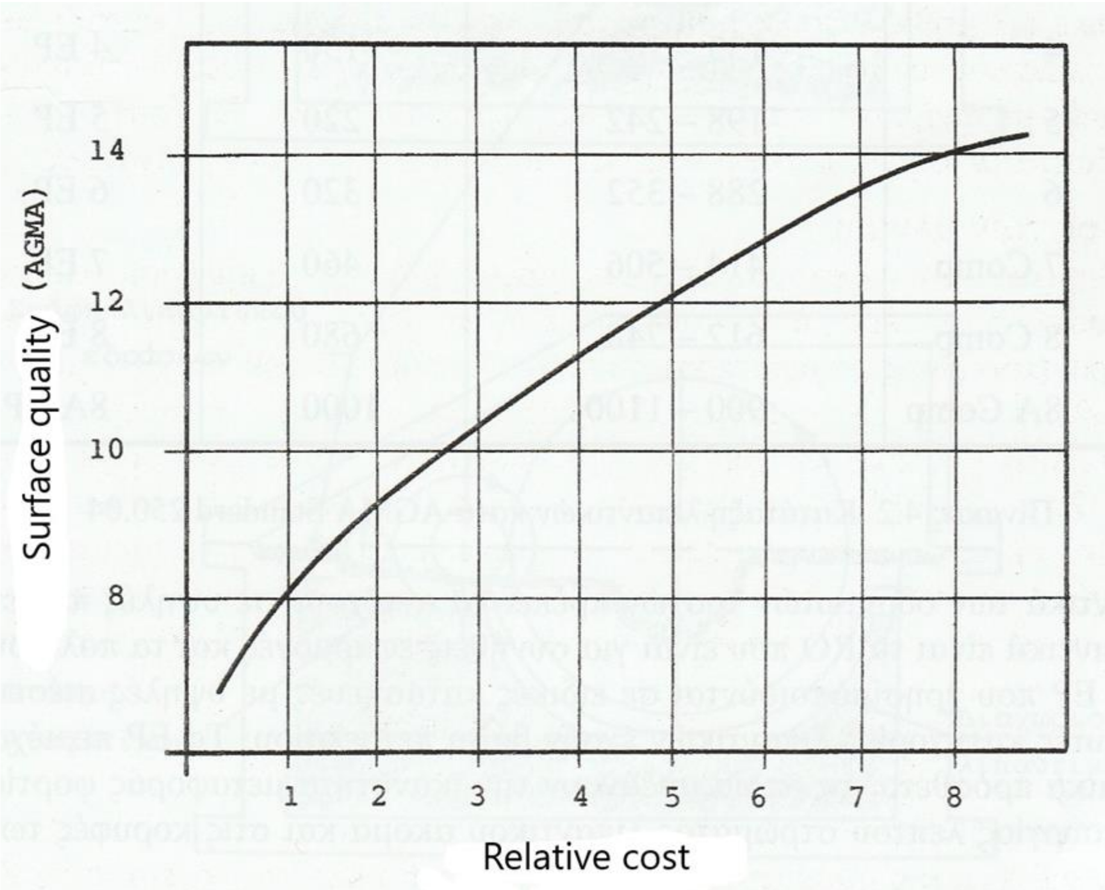
Cost of construction

This depends on the surface quality of the profiles. According to AGMA, the quality increases as the accuracy requirements increase (e.g., quality 8 is less accurate than quality 14), and the quality of the surface achieved is determined mainly by the processes of cutting and quality upgrading of the surface. Quality 8 can be used for low-speed wheels, while quality 14 can be used for high-speed wheels. Qualities 10-12 are suitable for cogwheels of vehicles and cars, as the accuracy requirements increase and the surface quality increases and obviously the construction cost increases, as in the next figure, where it seems that in the area of high quality a small increase in quality requires a large cost increase.

Surface quality by AGMA	Process
8-12	Hobbing, Shaving
9-13	Shaving
10-14	Grinding

The cost of trade materials

The high-quality construction of the wheel profiles must be accompanied by high quality materials from the trade. Damage to bearings and lubrication systems is usually more common than damage to wheels.



Sketch 4.12 Approximate relation between surface quality according to AGMA and construction cost.

Gear lubrication systems with dentations

In a gear unit the purpose of the lubrication system is to lubricate the power loss positions to reduce friction and remove the heat generated, which it transfers to the environment so that the temperature of the lubricant remains within the prescribed range. The positions of power losses, due to friction, are mainly the contacts of the cooperating profiles and the bearings of the shafts. The appropriate layer of lubricant must be maintained for good lubrication in these places.

There are various lubricants that can be used. Table 4.2 shows the standard (RO) lubricants and very high pressure (PE) lubricants used in sprockets.

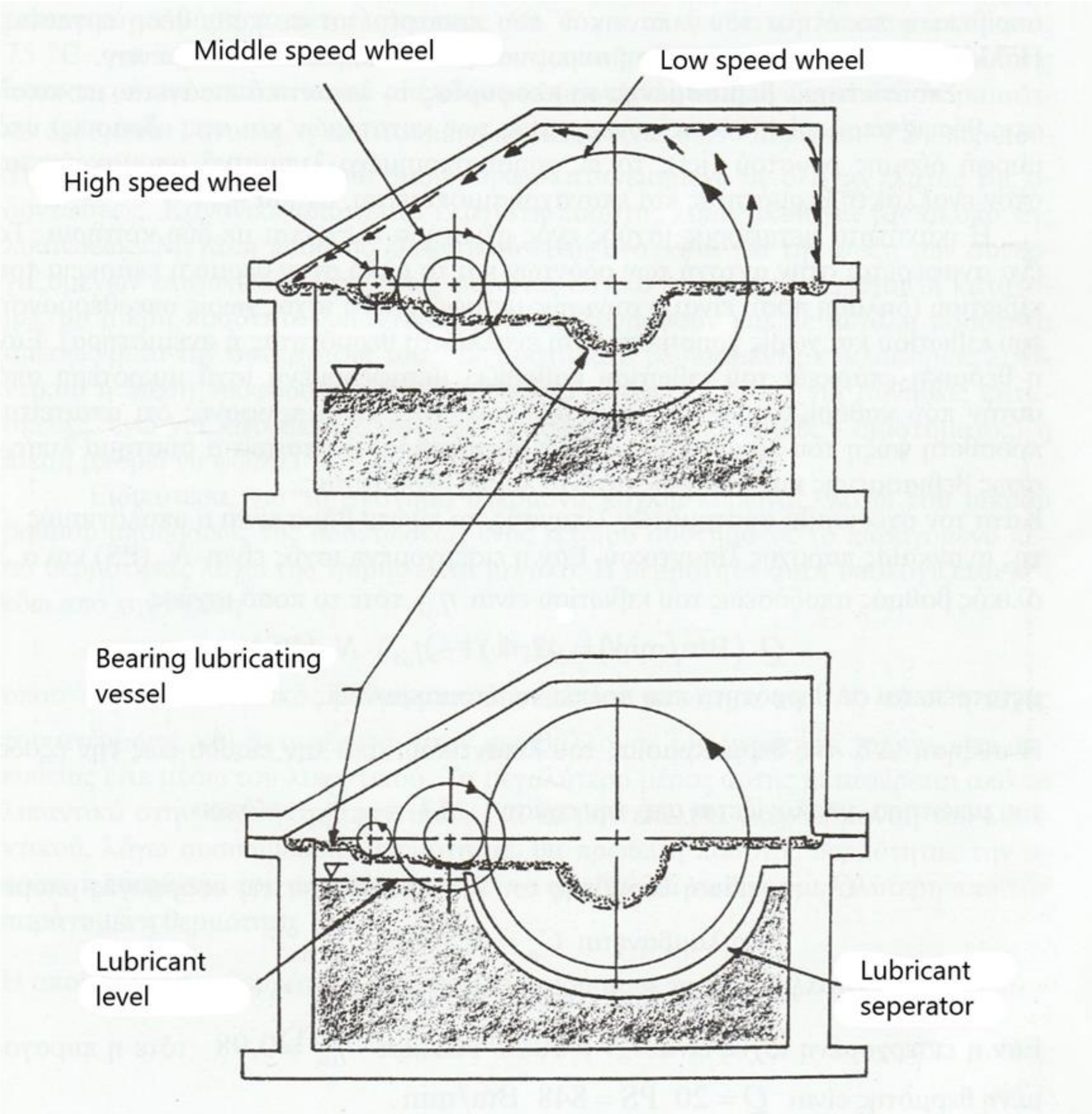
RO Lubricants AGMA number	Kinematic coherence mm^2/sec cSt in 40 ºF	ISO Classification	EP lubricants AGMA number
1	41.4-50.6	46	
2	61.2-74.8	68	2 EP
3	90-110	100	3 EP
4	135-165	150	4 EP
5	198-242	220	5 EP
6	288-352	320	6 EP
7 comp	414-506	460	7 EP
8 comp	612-748	680	8 EP
8A comp	900-1100	1000	8A EP

Table 4.2 classification of lubricants by AGMA Standard 250.04

Gear lubricants must be able to withstand high pressures. Such lubricants are the RO which are for common applications and the very high pressures EP used in special constructions with high pressures. Both of these lubricant categories are oil based. EP contain special chemical additives, which increase the ability to transport loads by creating a thin layer of lubricant even on the tops of the rough.

In addition to the above lubricants, there are also synthetic lubricants that come from chemical processes rather than oil. Compared to other lubricants they have some advantages such as being stable at high temperatures, having stable consistency in a large field of temperature change, having a long service life and can work efficiently at low temperatures.

These lubricants were created for military applications and usually carry military specifications. The synthetic lubricants used in the gearboxes of helicopters and gas turbines of airplanes have the ability to work throughout their operating cycle, at temperatures -50ºC to +200ºC. Lubricants Mil-L-23699 and Mil-L-7808, based on esters that are widely used are mentioned.



Sketch 4.13 Lubrication system with immersion

There are two types of lubrication systems, the immersion system and the hasty circulation system.

In the immersion system the tank contains lubricant up to a predetermined level, which is such that the wheel immersed in the lubricant reaches a depth of 4 to 6 standard modules below the lubricant level. The used lubricant returns to the tank. Figure 4.13 shows two immersion systems. The immersion system is simple and inexpensive but is limited to low speed and load reducers. As the velocities increase the heat generated is large and a heat exchanger is required to cool the lubricant. It is obvious that the lubricant must protect hard-to-reach areas, so the amount of lubricant offered at each job site must be precisely determined.

Properly positioned nozzles are often required.

In the forced circulation system, the lubricant is introduced by pressure in the loss positions (cooperation positions of the profiles and the bearings) in the form of a fluid bond and the used lubricant is removed to the heat exchanger and reused.

The power transmission capacity of a gearbox is tested by two criteria. One refers to the strength of the teeth and the other to the thermal adequacy of the box (i.e., how much is the continuously transferred power without overheating of the box and without the use of heat exchanger or fan). If the thermal adequacy of the gearbox determines a transmitted power less than that determined by the strength of the teeth, then it is obvious that additional cooling of the lubricant is required (e.g., with a fan) or a system of accelerated circulation lubrication with heat exchanger is required.

While designing the lubricating systems, the first step is the calculation of the required lubricant supply. If the incoming power is N (PS) and the total degree of efficiency of the gearbox is $\eta_{o\lambda}$, then the amount of power

$$Q(Btu/min) = 42{,}4 \cdot (1 - \eta_{o\lambda}) \cdot N\ (PS)$$

Is converted into heat that has to be removed.

The increase ΔT of the lubricant's temperature, from the input to the output of the gear, is calculated from the relation $\Delta T = \frac{Q}{\dot{V} \cdot C_p}$ where

C_p = Special heat of the lubricant, for the applications can be taken

$C_p = 0.5\ Btu\ /lb_m$°F

$\dot{V}$ = lubricant supply (lb_m/min)

If the incoming power is N = 1000 PS and $\eta_{o\lambda}$ = 0.98 then the produced heat is Q = 20 PS = 848 Btu/min.

If the lubricant supply is

$$\dot{V} = 2.5 \text{ lb/sec} = 150 \text{ lb/min}$$

Then the increase of its temperature will be

$$\Delta T = \frac{848\ Btu/min}{150\ lb_m/min \cdot 0.5\ Btu\ /lb_m °\text{F}} = 11.3\ °\text{F}$$

It is obvious that some gear points will have temperatures much higher than the lubricant temperature. The maximum temperature in the gearbox must not exceed 50ºC for oil-based lubricants, 60ºC for synthetic lubricants and 75ºC for very good gas turbine gearbox lubricants. The amount of lubricant required in a denture is usually calculated from experimental data. Generally, the quantity required is about 0.017 lb./ (min·PS). The lubricant must be evenly distributed over the entire width of the dentition. Normally a minimum amount of lubricant is required for the purpose of lubrication. A large amount of lubricant is necessary for the cooling of the cooperating surfaces. Therefore, it seems reasonable to supply the profiles with a small amount of lubricant before they cooperate and with a large amount immediately after their cooperation. In systems of forced lubricant circulation, the pressure or supply pressure at the nozzles reaches 2 atu for ordinary constructions, while for constructions of high demands (aviation-space) the pressure can reach 7 atu.

In particular, for the endless screw-crown system due to the low degree of denting efficiency of such a system the amount of heat generated due to friction is large. This heat is also calculated here from the relation:

$$Q = (1 - \eta_{o\lambda})N_1$$

Where $\eta_{o\lambda}$ the total efficiency of the gearbox and N_1 is the power of the worm, and is transferred to the surrounding gearbox space, it is directly through the lubricant. Most of it is transported from the lubricant to the lubricant storage. In order to prevent overheating of the lubricant, due to heat accumulation, the amount of heat that the surface of the reducer can eliminate must be greater than the heat Q produced.

The eliminated from the gear to the environment heat is

$$Q' = h_t \cdot (T_{max} - T_0) \cdot A \cdot (1 + \zeta)$$

Sketch 4.14 shows an accelerated lubricating system in which the following are distinguished:

A = lubricating pump, from the gearbox shaft

B = lubricant filter

C = heat exchanger (cooler)

D = lubricant tank

E = manometer

F = thermometer

G = check valve

H = pressure regulating valve for by-pass creation

I = lubricant suction pipe

K = lubricant discharge tube

L = lubricant return pipe

M = lubricant level indicator

N = starter (before pump's A operation) and auxiliary pump

O = flow and temperature calibrator

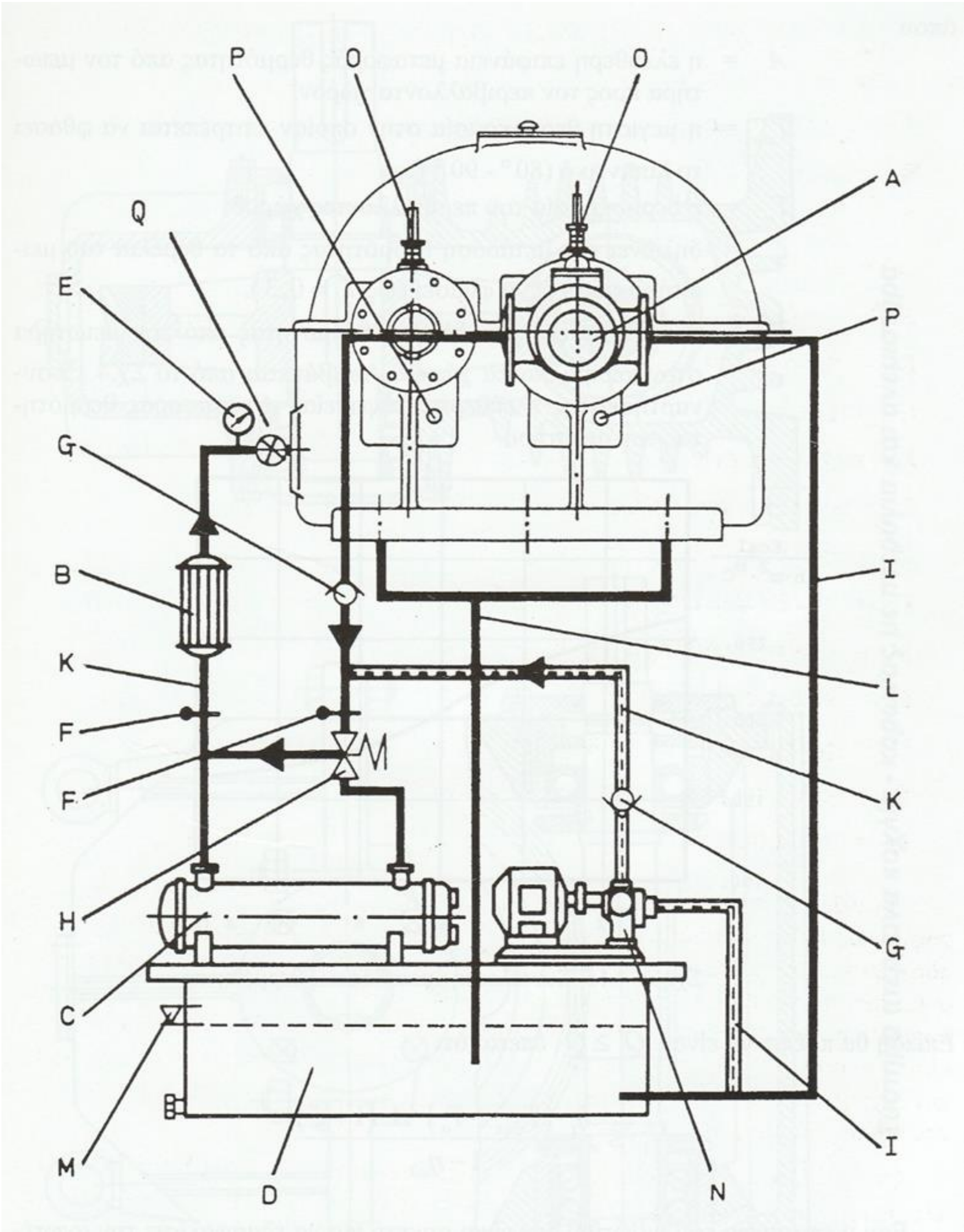
P = connection for temperature measurement

Q = pressure regulator

Pump A sucks lubricant from tank D through the section pipe I. In the discharge the lubricant is led to the heat exchanger C through the check valve G and through the filter B reaches the pressure regulator Q. The pressure regulator is located at the input of the lubricant to the gearbox, in order to retain the pressure at a predetermined value. The auxiliary pump N is used for lubrication before the gear operates and is also the system's backup in case of failure of the main pump A.

Check valves G are placed at such places of the lubrication circuit so that the main pump A does not derive through the piping of the auxiliary pump N and the discharging pipe of the main pump A cannot channel lubricant to the auxiliary (secondary) pump N.

By-pass H (which is thermostatically regulated) regulates the lubricant supply to the cooler C, so that the lubricant will not be cooled at low temperatures. Thermometers and manometers are placed in various positions of the lubricating circuit.



Sketch 4.14 Accelerated circulation lubrication system

Where

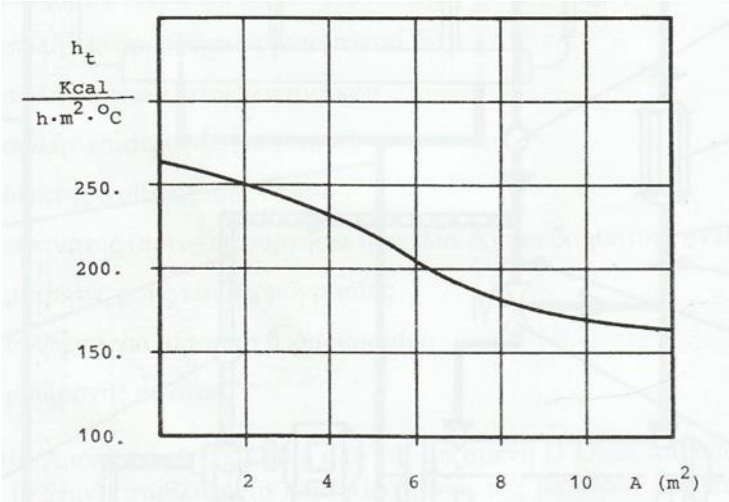
A = the free heat transfer surface from the gear to the surrounding area.

T_{max} = the maximum temperature that the lubricant is allowed to reach (80º-90º).

T_0 = the temperature of the surrounding area.

ζ = indicates heat transmission from the foundations of the gear (it can reach up to $\zeta = 0.3$)

h_t = the heat transfer factor from the gear to the surrounding area. It is taken from sketch 4.15 as a function of the free heat transfer surface A of the gear.

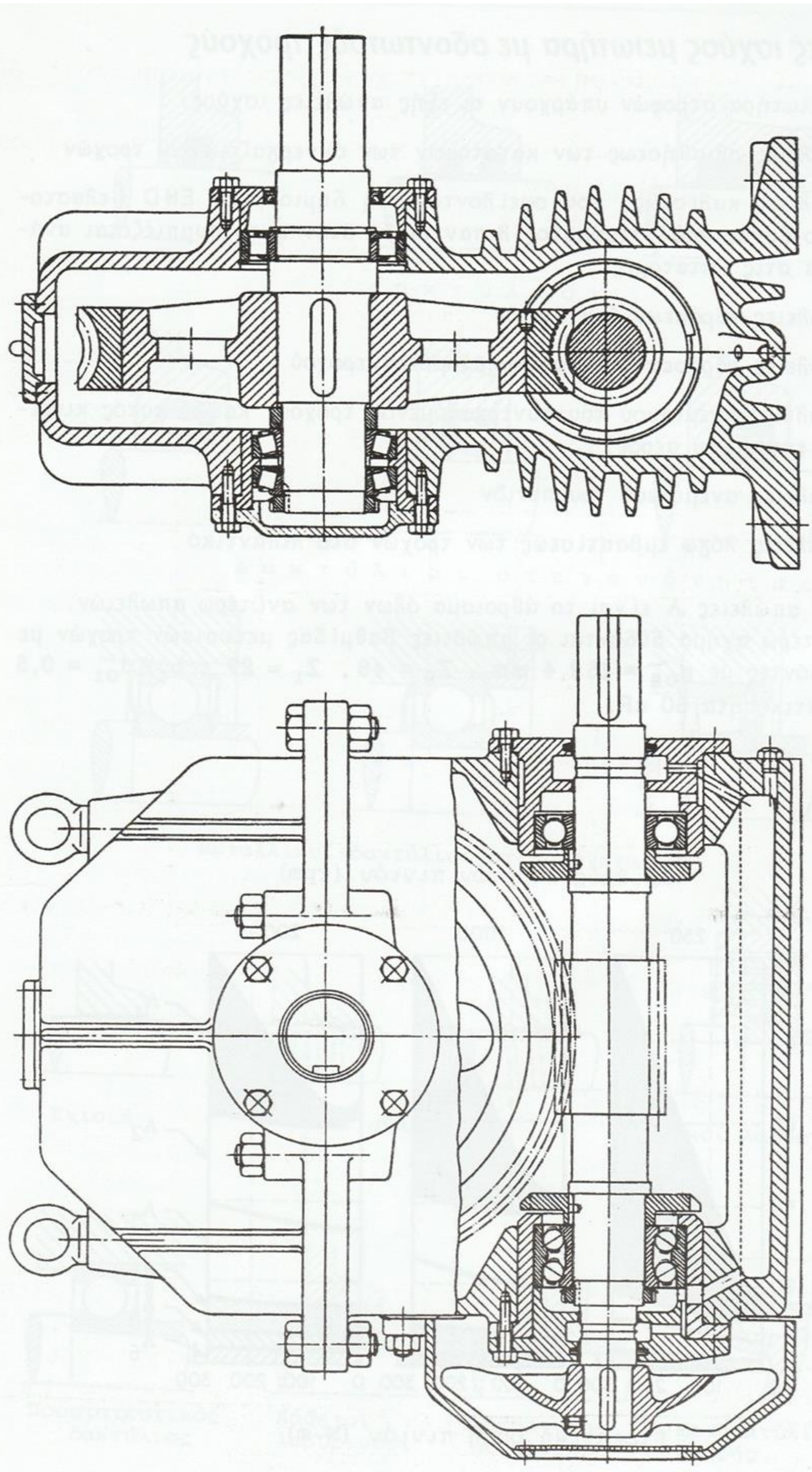


Sketch 4.15 Coefficient of heat transfer

Whereas $Q' \geq Q$ must be in valid, it follows that

$$N_1 \leq \frac{h_t \cdot (T_{max} - T_0) \cdot A \cdot (1 + \zeta)}{1 - \eta_{o\lambda}}$$

If the surface of the gearbox is not enough to ensure the relation above, wings are constructed (parallel to the cooling air flow current) which can increase the heat dissipation surface up to 50%. An increase of the dissipated heat amount is accomplished with a strong ventilation through a fan and further through a heat exchanger at the return line of the lubricant, as in the following figure.



Worm gear-crown system with blades and fan

Power losses of a cogwheel gear.

In a gear unit there are the following power losses:

A_1 = sliding losses of the profiles of the cooperating cogwheels

A_2 = rolling losses, that are due to the creation of EHD (elastic-hydro-dynamic) lubricant layer, when it is compressed between the profiles.

A_3 = bearing losses of the pinion

A_4 = bearing losses of the cooperating wheel

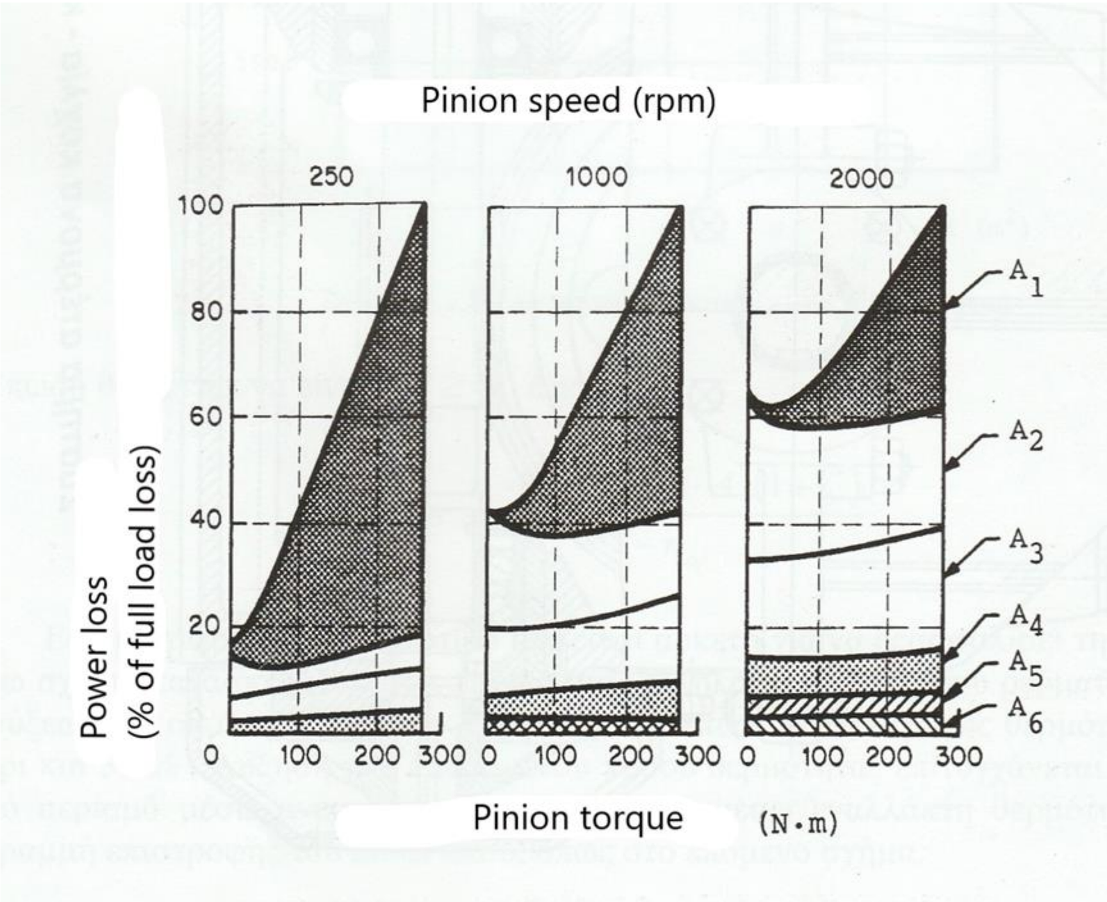
A_5 = wind losses of cooperating wheel, while moves in the air

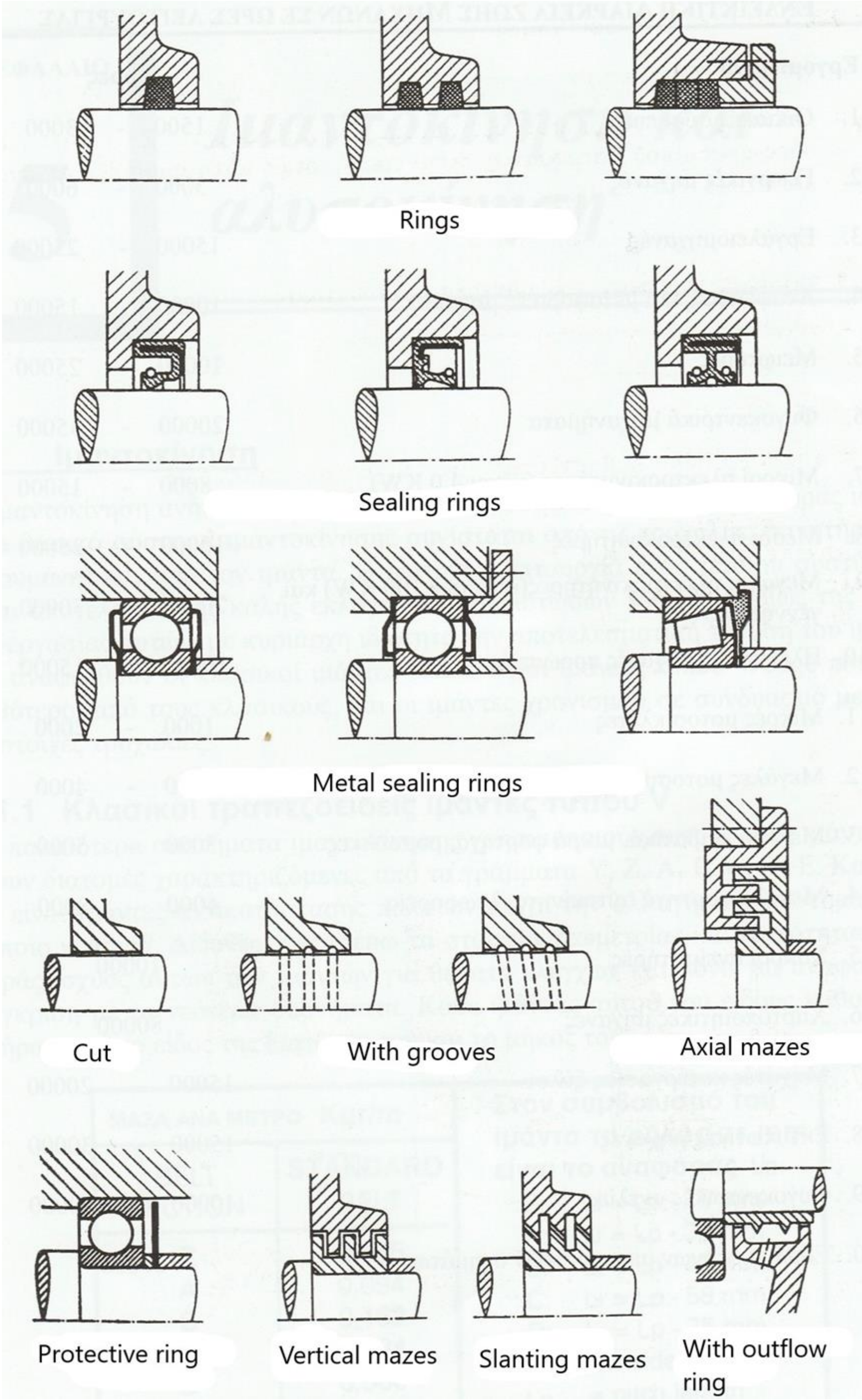
A_6 = wind losses of the pinion

A_7 = losses due to the wheels' lubricant immersion.

Total losses A are the summary of the whole losses above.

Frontal wheel step losses with straight teeth with d_{02} = 152.4 mm, z_2 = 48, z_1 = 29, b_1/d_{01} 0.5 and coherence 30 cP are given in the sketch below.





Methods of gear shaft sealing

INDICATIVE LIFE SPAN OF MACHINES IN OPERATING HOURS

Construction machine	hours
1. Home devices	1500-3000
2. Agricultural machines	3000-6000
3. Machine tools	15000-25000
4. Lifting and transporting machines	10000-15000
5. Gears	10000-25000
6. Centrifugal machines	20000-35000
7. Small electric engines (up to 4.0 KW)	8000-15000
8. Average engines	15000-25000
9. Large electric engines (above 100 KW) and generators	20000-30000
10. Electric propulsion machines	10000-15000
11. Small motorcycles	1000-2000
12. Large motorcycles, small cars	2000-4000
13. Large passaging, small trucks, trailers	3000-5000
14. Large trucks, buses	4000-8000
15. Small fans	10000
16. Papering machines	80000
17. Woodworking machines	15000-20000
18. Printing machines	15000-30000
19. Centrifugal pumps	10000-30000

20. Axes bearings of transporting vehicles	20000

CHAPTER 6: HYDRAULIC SYSTEMS

6.1 SIZES AND DEFINITIONS

In hydraulic systems measurements rely on the static pressure, because this takes large degrees and because so the altitude differences as the flow velocities are of little importance for the calculation of the total pressure of the fluid. Consequently, the pump gives the fluid theoretical power

$$N_{p,th} = \delta p \cdot \dot{V}_{th}$$

Where δp = the increase of the static pressure (due to low speeds and despite the possible difference of speeds in the suction and discharge branches).

$\dot{V}_{th}$ = the theoretical supply (the dot shows the reduction of supply in the time unit).

If n the pump's number of rotates per unit of time, then the angular velocity will be $\omega = 2\pi\, n$, and if

$V =$ The theoretical pump supply per rotation and width unit

V_p = the theoretical pump supply per rad of rotation and

b = the width of the pump

With a combination of the forms and definitions above, we can easily conclude that,

$$\dot{V}_{th} = \omega V_p = 2\pi\, nV_p = n\left(2\pi V_p\right) = nbV$$

$V_p = b\,\frac{V}{2\pi}$ Where $V = \int_0^{2\pi}\left(\frac{dV}{d\theta}\right)\delta\theta$

Where $dV/d\theta$ the instantaneous supply of the pump per unit of width and rotation rad.

Therefore,

$$N_{p,th} = \delta p\ \omega\ V_p$$

The theoretical torsional torque of the pump $M_{d,p,th}$ is such that

$$N_{p,th} = \omega\ M_{d,p,th}$$

And from the combination of (1) and (2) results

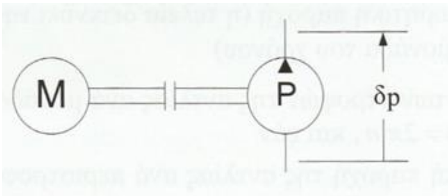
$$M_{d,p,th} = \delta p\ V_p$$

In the engine-pump system of the sketch

N_K = power consumption from the engine for the pump movement

$M_{d,K}$ = the developing torque to the engine = N_K/ω_K

$\omega_K = \omega_p = \omega$ = angular velocities of engine and pump, which are the same due to fixed connection of the engine's and pump's shaft.



Sketch 6.1 Pump-engine system and increase of static pressure

The ratio of the theoretical power $N_{p,th}$ to engine power N_K, is called mechanical efficiency degree η_m of the system,

$$\eta_m = \frac{N_{p,th}}{N_K} = \frac{M_{d,p,th}}{M_{d,K}} = \delta p \frac{V_p}{M_{d,K}}$$

Where, as above,

$$M_{d,K} = \frac{N_K}{\omega} = 71620 \frac{N_K(PS)}{n_K(RPM)}$$

For the common hydraulic systems can be taken, with insurance,

$$\eta_m = 0{,}85 \div 0{,}95$$

The ratio of the real supply $\dot{V}$ to the theoretical supply $\dot{V}_{th}$ of the pump, is called volumetric degree of efficiency η_Y,

$$\eta_Y = \frac{\dot{V}}{\dot{V}_{th}} = \frac{\dot{V}}{\omega V_p} = \omega \frac{V_{p,\pi\rho}}{\omega V_p}$$

Which is calculated based on the hydraulic losses in the pump and for the hydraulic systems, it usually has rates $\eta_Y = 0{,}80 \div 0{,}90$.

The real rate of pump supply per rotation rad is

$$V_{p,\pi\rho} = \eta_Y V_p$$

And

$$M_{d,p,th} = \delta p V_p$$

In addition,

$$N_{p,th} = M_{d,p,th}\omega = \delta p \omega V_p = \delta p\ \omega \frac{V_{p,\pi\rho}}{\eta_Y}$$

Or

$N_{p,th} = \frac{N_{p,\pi\rho}}{\eta_Y}$

Where

$N_{p,\pi\rho} = \delta p\ \dot{V} = \delta p\ \omega\ V_{p,\pi\rho} = N_p,$

is the power actually delivered in the working means.

From above

$$\eta_m = \frac{N_{p,th}}{N_K} = \frac{N_p}{\eta_Y N_K}$$

And

$$\eta_m \eta_Y = \eta_{o\lambda} = \frac{N_p}{N_K}$$

The last relation was expected due to the definition of the total efficiency degree of the system, since it is generally true that the ratio of the final power of the consumed power is the total efficiency degree of a system.

It is obvious from the above that the dimension V_p of the pump, that is the theoretical pump supply per rotation rad, plays a significant role to the calculation of other sizes of the system. V_p is clearly a geometrical size and clearly depends on the type and size of the pump. Next follows an in-depth research for the calculation of this so important size, because it is clear that knowledge of this calculation enables the designer or even the manufacturer to know the specifics of the pump, as well as its performance at each step while functioning. Many times, **special supply,** q_p **,** is met at the international bibliography, and also the manufacturers prefer it, which is the provided fluid volume per rotation of the pump calculating as:

$$q_p = 2\pi V_p$$

Example 6.1

Hydraulic pump has special supply $q_p = 10\pi$ (cm^3), shows real power to the fluid $N_p = 12.5$(kW), creates $\delta p = 150$bar and operates at 1800RPM. The engine of the pump consumes 15kW. (1kp=9.8N). Asked to be calculated:

1) The mechanic efficiency degree of the engine-pump system.

$$\eta_m = \frac{M_{d,p,th}}{M_{d,K}}$$

Theoretical torsional torque of the pump, ($V_p = 5$ cm^3/rad),

$M_{d,p,th} = \delta p V_p$ = 150 × 5 (bar·cm^3) = 765 kp · cm

Torsional torque of the engine

$M_{d,K}$ = 71620 × 15 × 1, 36 (PS) / 1800 (RPM) = 812 kp · cm

So,

$\eta_m = \frac{765}{812} = 0{,}942$ (Within the prescribed limit)

2) *Hydraulic losses*.

Total efficiency degree

$$\eta_{o\lambda} = \frac{12{,}5}{15} = 0{,}833$$

Volumetric efficiency degree

$$\eta_Y = \frac{\eta_{o\lambda}}{\eta_m} = \frac{0{,}833}{0{,}942} = 0{,}884$$

Average theoretical supply

$\dot{V}_{th} = \omega V_p = 2\pi \times 1800 \times \frac{5}{60} = 942$ cm^3/s = 9.42 × 10^{-4} m^3/s

Real supply

$\dot{V} = \eta_Y \, \dot{V}_{th} = 0{,}844 \times 9{,}42 \times 10^{-4}$ m^3/ s

So the hydraulic losses will be

$\delta\dot{V} = \dot{V}_{th} - \dot{V} = (1 - \eta_Y)\dot{V}_{th} = 1{,}09 \times 10^{-4}$ m^3/ s.

6.2 POSITIVE DISPLACEMENT PUMPS TECHNOLOGY

Hydraulic pumps convert mechanical power, which derives from their turbine, in hydraulic power of the cooperating means, the activation of which is the same in all types of pumps.

In all pumps the moving parts create a vacuum in the suction in the sense that the fluid can insert from the suction pipe to the pump. In addition, they create inseparably of the fluid in suction in the sense that the fluid, not possible to return to the suction, is pushed to the discharge tube. Pumping is developing in that way, which is the efficient operation of the pump. Although principle of pumping is the same, yet the pumping components are different in the various kinds of pumps. On industrial hydraulic systems pumps with cogwheels are usually used, with wings and pistons and more rarely with rotating or not primary fluid bundles.

All kinds of pumps share some common characteristics. They both require an appropriate supply of aspirated fluid volume for the allocation of the desirable fluid supplement on the installed systems, with the correct pressure. Therefore, all of them are affected from the various natural properties of the fluids (such as density and coherence) and the flow characteristics.

Positive displacement pumps are designed to displace a determined volume of fluid every time an operating circle is completed. This results in not to be prone to the various operating situations to the system that follows the pump and leads to flow. However, there are some exceptions. Some types of positive displacement pumps, which in their mode of operation like the bolts, are extremely sensitive to the fluctuations of the operation point from the pressure to the depression.

When positive displacement pumps are used, the system receiving the activated fluid must be protected against extreme pressure values. The pump supplies fluid to the discharge at any pressure required to overcome the load. The only limitations on the maximum pressure that develops are the pressure that destroys the components of the equipment and the maximum value of the pressure that can be produced by the engine-pump system.

Due to the possibility of developing almost unlimited values, all positive displacement pumps must have relief valves on the discharge side where the flow is directed. This one is required to protect the pump and piping from overpressure. Some system designs include a relief valve located inside the pump hull. Other types of systems use a separate valve assembled at the beginning of the discharge.

Positive displacement pumps are capable of providing a constant supply of the fluid to their output for each of their operating cycles. Therefore, the only factor that affects the flow rates in an ideal positive displacement pump, is its rotational speed. The flow resistance in the system in which the pump operates does not affect its flow rate. The actual flow of the positive displacement pump is less than its theoretical flow and as the discharge pressure of the pump increases the fluid escapes as an internal leak from the discharge to the suction.

There is a minimum necessary grace that is required for the proper operation of the pump, but it must at the same time be such as to minimize the damages. Proper and careful operation and maintenance of positive displacement pumps minimize the amount of the internal leaks.

Positive displacement pumps are produced in a wide variety of configurations and formations. Each configuration has a separate function and can be preferred based on its efficiency and reliability in the specific application for which it is intended.

The main characteristics of the hydraulic pumps are:

- ✓ The *maximum continuous operating pressure*,p_{max} , which is the maximum pressure at which the pump is allowed to operate continuously. The life of the pump is inversely proportional to the operating pressure. If a pump is operating at pressures higher than p_{max} then its service life is reduced mainly because the pump bearings are affected by high pressures. Structural improvements such as the use of hydrostatic (with pressure) sliding bearings or balancing plates to balance the loads of the bearings, allow an increase ofp_{max}.
- ✓ The *maximum short-term pressure* p'_{max} that is greater than pressure p_{max} and at which it is possible for a pump to operate for a short time. The duration of the overpressure and the frequency of its appliance are determined by the manufacturer.
- ✓ The maximum number of continuous operating turns, n_{max}.it has been shown that the number of operating turns and the life of the pump are amounts inversely proportional. The maximum number of turns is also determined by the risk of cavities in the pump. The position of the working

means tank (lower or higher than the pump) also affects the maximum number of turns of continuous operation of the pump.

Other characteristic sizes of the pumps have already been mentioned before, which the person dealing with the hydraulic systems must keep in mind. It must also be known whether the pump will be fixed or variable (adjustable) flow and provided by the manufacturer various technical elements, such as the inertial torques of the rotating parts, the specific flow, the efficiency degrees, the operating temperature area of the pump, the suitable fluid or the viscosity of the working means and other different elements that refer to the purity of the fluid and the particular characteristics of the pump.

If

n_p = number of rotates of the pump in RPM

$n_{m,p}$ = mechanical efficiency degree of the pump

$n_{Y,p}$ Or $\eta_{Q,p}$ = volumetric efficiency degree

$\eta_{o\lambda}$ = total efficiency degree

q_π = special pump supply in cm^3/speed

$\dot{V}_p$ = efficient flow under the pump in lit/min

N_p = power consumption in kW

δp = pressure difference from the discharge to the suction at bar

$M_{d,p}$= torsional torque at the shaft of the pump at Nm.

And having in mind what has been mentioned at the beginning of the chapter, we can write the following:

Efficient supply

$\dot{V}_p$(Lit/min) = $q_p n_p \eta_{Q,p} \times 10^{-3}$

Consumed power

N_p(KW) = $\frac{M_{d,p} n_p}{9549} = \frac{\dot{V}_p \delta p}{600 \eta_{o\lambda}}$

Torsion torque at the pump shaft

$M_{d,p}$(N·m) = $\frac{1.59}{100} \frac{\dot{V}_p \delta p}{\eta_{m,p}}$

The pump does not create pressure. Its job is to move fluids. During its operation it receives the fluid from the suction and pushes it to discharge. The pump liquid, as it is pushed into the discharge, encounters resistances in its flow, with each rotation angle, which creates the pressure.

The ever-increasing use of the pumps and generally of hydraulic systems, in the operation and automation of machines and devices for transporting and handling materials and products is due to their advantages, which include the smoothness of their operation, their ease appliance and adjustability and that they are relatively easy and simple to use.

Pumps are designed with positive internal sealing to prevent the return of the fluid to the suction, which is from the high pressure to the low-pressure space. As this cannot be completely avoided, there is a small return of fluid to the suction that increases as the pressure at discharge increases. Thus, as the pressure in the discharge increases, the pump supply decreases.

Depending on the appliance, a hydraulic system requires a fixed or variable supply flow pump. The change of supply is achieved in two ways:

- By changing the speed of the pump shaft
- By changing the supply per rad of the pump, as the speed will be constant.

In constant supply pumps the supply can only be changed through speed change. This type of pumps is usually used in hydraulic systems in which the supply is almost constant or changes slightly and are simpler in their use than the variable supply pumps.

In hydraulic systems with variable supply requirements such a pump shows its advantages. Such a pump rotates at constant speeds and is constructed in such a way that its supply can take any rate from zero to maximum possible.

A suitable device that changes the geometry of the pump achieves this mechanically. As variable supply pumps are more complicated from the constant flow pumps, their use is restricted in appliances that require change of the supply indeed.

The ability of the changing supply pumps to regulate their supply depending on the requirements of the hydraulic system allows the user to reduce the energy consumption, abstract some valves from the system and decrease the problems of overheating of the working means. Although variable supply pumps have a higher initial cost and require more maintenance, yet the advantages above contribute to their increasing rate use.

The operation of positive displacement pumps is affected by three fundamental factors: fluid coherence, rotating speed and fluid supply at suction.

Coherence. Positive displacement pumps are designed to handle cohesive liquids, such as oils, greases and polymers. However, a change in fluid coherence exerts important affection at the operation of the pump. While the coherence increases, the pump should operate by absorbing more power from the engine for the sustenance of constant operating speeds, in order to satisfy the desirable supply of activated fluid at the output. If the coherence increases excessively then there will be a value beyond which the power required to move the fluid is greater than that offered by the engine.

The variety of operating temperatures is an important factor as it directly affects the coherence of the moving fluid. The design specifications of the pump must determine the operating temperature ranges that are allowed, as well as the range of coherence. These two parameters are inextricably connected and must have been carefully studied.

Rotation speed. In positive displacement pumps the output is directly proportional of the rotation speed. If the speed changes then the operating point also moves from its normal position and the volume of the moving fluid, that is the supply, changes too.

Suction fluid supply. Positive displacement pumps can absorb the fluid and lead it to the suction chamber, to some extent. However, a steady supply of feed fluid must be available. Therefore, the suction system of the pump must ensure an available constant non-turbulent fluid supply at its input.

The operation of the pump and its useful lifetime increases if the suction system inserts into it fluid at constant pressure. When pumps work harder to achieve the suction of the fluid, it is difficult for them to depress it.

6.3 GEAR PUMPS WITH EXTERNAL GEAR

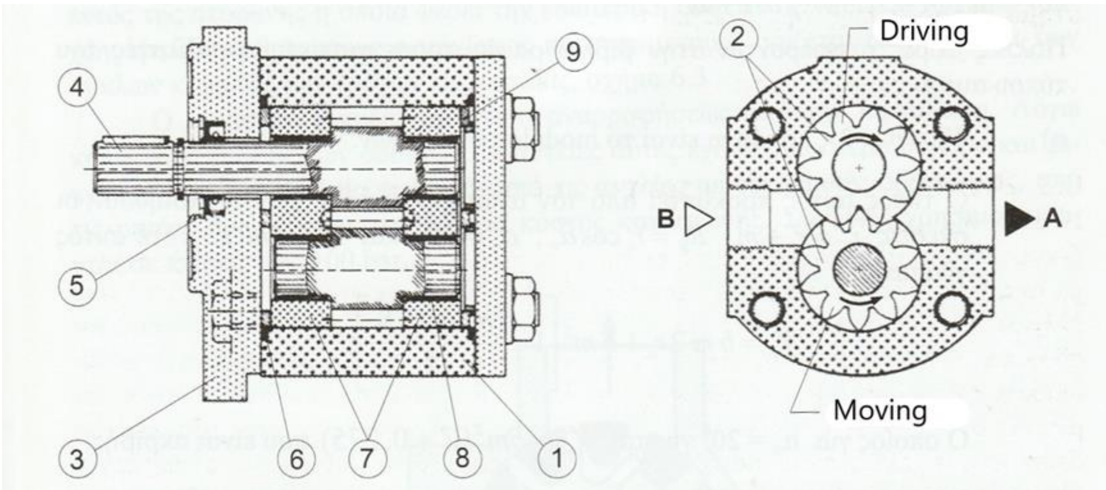
The simple gear pump of front toothed wheels consists of two front toothed wheels that rotate in opposite directions by cooperating in the same hull. There is a small grace of the order of few μm between the hull, the faces of the wheels and at the edges of the teeth. This design pushes the fluid to fill the space that is bounded from two consecutive teeth and the hull and move with them as the wheels rotate. When the teeth of one wheel are coupled with the teeth of the other, the space between them decreases. This creates in inseparable in the area of discharge and the trapped fluid is led to depression.

As the cogwheels rotate and the teeth uncouple, the space reopens at the side of the pump suction and traps new amounts of fluid that transfers regional to the hull of the pump to the depression. As the liquid fends off the suction area, the pressure reduces and new liquid is aspirated at the suction area of the pump.

In case the pump gears have a large number of teeth, depression presents smoother and more continuous flow of fluid, since small amounts of liquid follow each other in a short time. The space between the teeth is larger and so for given rotation speed the supply that moves is larger for cogwheels with a smaller number of teeth. However, this increases the tendency for fluid to circulate in depression.

In a simple front wheel pump, the power of the engine is applied to rotate one of the two shafts on which the drive gear is assembled. The second wheel gets motion through the coupling of its teeth and its cooperation with the drive wheel. There are no valves inside the gear pumps that cause friction losses, as in reciprocating pumps. High speeds imposed on the impeller of the centrifugal pumps and has as a result the increased energy losses due to friction, are not necessary in gear pumps. This makes gear pumps more appropriate for cohesive liquids, like fuels and lubricants.

Sketch 6.2 shows the components of such a pump. While the drive wheel rotates in cooperation with the moving, the process of successive uncoupling of the teeth at suction B and their coupling at depression A creates increased volume in suction and decreased volume in depression, thus developing the pumping mechanism.



Sketch 6.2 Standard pump with external sprockets.

1. The hull
2. The drive and moving wheel
3. The lid

4. The drive shafts
5. The seal of the shaft
6. The sealing rings
7. The bearings
8. The bearing rings for high loads
9. The hydrostatic load balancing

The fluid inserts and occupies the available space in suction. It is then transferred regionally under the teeth to the depression where the teeth begin to couple occupying a portion of the available space and thus the fluid is forced to proceed to the downstream of the depression flow.

The special supply ranges from 3.5 to 100 cm^3/speed and operation pressure is usually up to 250 bar. Cogwheels are made of steel and there is a special hardening with surface paint. The lateral friction surfaces are made of phosphorous brass or a special alloy.

The external gear pump supply per rad will be

$$V_p = b\left[r_k^2 - r_o^2 - \frac{\left(\pi R_g\right)^2}{3Z^2}\right] = \frac{q_p}{2\pi}$$

Where q_p the special supply

Approximate variants of that form are many times referred in bibliography

a) $V_p = b\ m\ 2r_o$, where m the module of the wheels.

This form derives from the above provided that the relations

$r_k = r_o + m,\ R_g = r_o cos a_0,\ a_o = 20^0$ and $r_o = Z\ m/2$ are used. Then this becomes:

$$V_p = b\ m\ 2r_0 + bm^2\left[1 - \frac{(\pi cos_a)^2}{12}\right]$$

Which for $a_o = 20^0$ becomes $V_p = bm^2(Z + 0.275)$ which is accurate.

b) By omitting the second additive from the relation above, it results

$$V_p = b\ m\ 2r_0 = bZm^2$$

c) Another expression is the following:

$$V_p = 0.5\, b\, \left(r_k{}^2 - r_f{}^2\right)$$

That last form gives worst approach. If the previous relation and that $r_f = r_o - 1.25m$ are used, results that:

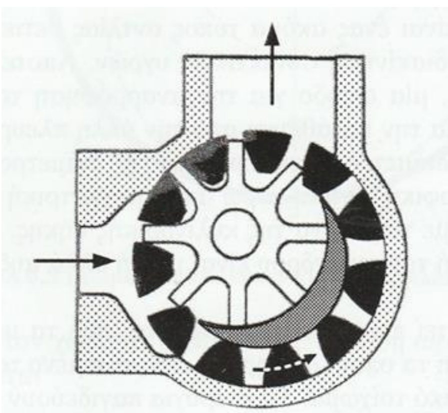
$$V_p = b\, m\, 2r_0 - b\, m\, r_o \frac{1}{4}\left(\frac{2.25}{2} - 1\right)$$

<u>Gear pumps with helical front wheels,</u> have several advantages over the pumps with front straight gear wheels. The last operate with their teeth coupled along their entire length at the same time. In helical gears, on the contrary, the contact point of the cooperating wheels moves along the length of each tooth, from one side to the other, on their working jowl, while the drive shaft rotates. This results in the depression having more stable pressure value and smaller, down to zero, supply pulse from the pumps with straight gear wheels.

6.4 GEAR PUMPS WITH INTERNAL GEAR

The wheel with the outer gear is the small one and is located inside the rim which carries the inner dentation. The meniscus, which is geometrically defined by the arcs of the head circles of the pump wheels, is placed between the two wheels.

Meniscus separates the aspiration area from the depression. Due to better contact of the teeth, these pumps have less noise and less flow unevenness than the pumps with external teeth, yet from which they have higher construction cost. They are usually used at pressures up to 100 bar.

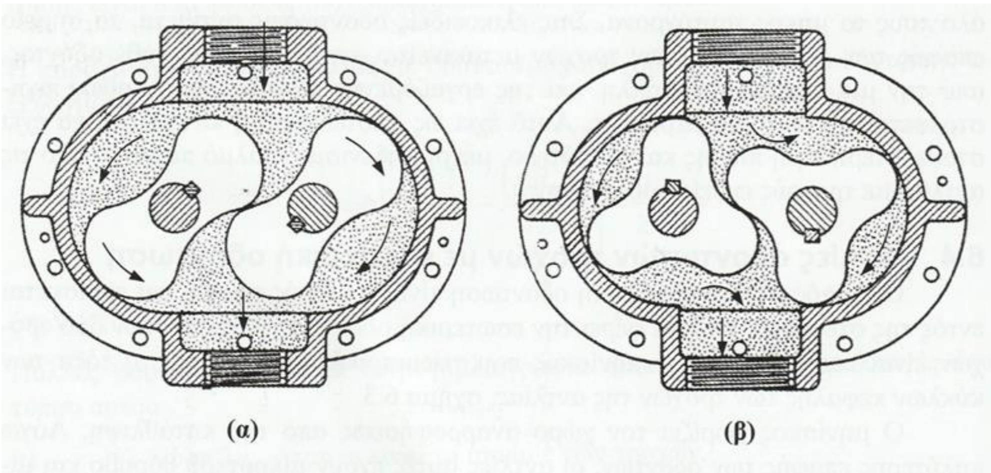


Sketch 6.3 Gear pump with internal gear.

6.5 PUMPS WITH PODS

Pump with pods is another form of simple gear pump. It can be considered as a simple gear pump with only two or three pods per wheel. Apart from this difference, its operating principle and the operation of all components that make it up, has no other differences.

They are the closed contact truck pumps. They work as the other cogwheels and the motion is given externally in both of them at the same time through a pair of advanced cogwheels, which are located on their shafts, sketch 6.4.



Sketch 6.4 Pump with (a) two and (b) three pods respectively.

6.6 FLAP PUMPS

Flap pump is one more kind of positive displacement pump and is used for the transport of cohesive liquids. It consists of a hull with a cylindrical hole, an input for fluid's suction on the one side and an output for depression on the other side. A cylindrical rotor with diameter smaller than the diameter of the cylindrical hull, is rotated about a central axial line which is eccentric about the axis of the cylindrical hull. The grace between the rotor and the top of the cylinder is small but increases as we move to the base.

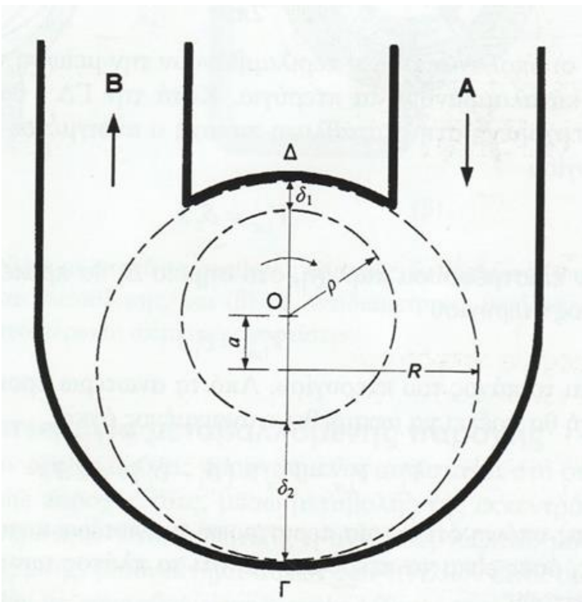
The rotor has flaps that move inwards and outwards as it rotates and that they keep the space between the rotor and the cylindrical wall isolated. Flaps trap fluid from the side of the pump suction and transfer it to the depression area where the narrowing of the space creates inseparable and throws the fluid to the depression pipe. Flaps can swing on an axis or slide on the slits of the rotor.

The cylinder carrying the flaps, which regress in suitable sockets that exist on the body of the cylinder, has a radius ρ, and rotates about O, which is eccentrically positioned in the cylinder with radius R. the heads of the flaps erase the circumference of the radius R, so that in the upper position only the length $\delta_1$of a flap protrudes, while in the lower position the length δ_2 of another flap protrudes. From the sketch below results:

$$\delta_1 = R - \rho - a \text{ And } \delta_2 = R - \rho + a$$

, where a is the eccentricity of the cylinder. Suction takes place from A and depression at B, for a given pump that is shown by sketch at the previous figure. According to the previous theory of supply, the inserting supply in depression on point C, per width unit of the flaps, for rotation angle $d\theta$ will be:

$$dV_{\varepsilon\iota\sigma.} = (a+R)^2 \frac{d\theta}{2} = (\rho + \delta_2)^2 \frac{d\theta}{2}$$



Sketch 6.5 Geometry and eccentricity of pump with blades.

And the returning to the suction area, that is the removable from depression supply will be:

$$dV_{\varepsilon\pi\iota\sigma.} = (\rho + \delta_1)^2 \frac{d\theta}{2} = (R - a)^2 \frac{d\theta}{2}$$

The difference between them gives the net supply of the pump at the output B,

$$dV = dV_{\varepsilon\iota\sigma.} - dV_{\varepsilon\pi\iota\sigma.}$$

So

$$\frac{dV}{d\theta} = 2\ a\ R$$

Supply per width unit and for a rotation of the pump, $0 \leq \theta \leq 2\pi$, will be

$$V = \int_0^{2\pi} \left(\frac{\delta V}{d\theta}\right) \delta\theta = 4\pi\ a\ R$$

Supply per rad, on consequence will be

$$V_p = b\frac{V}{2\pi} 2b\ a\ R$$

Up to here the calculations do not include the supply reduction due to the space that is occupied by the flaps. At CD, the flap volume opened in the unit should be deducted from the inbound supply

$$V_{1,\pi} = \delta_2 s$$

While from the returning supply at point D, the corresponding flap volume should be removed

$$V'_{1,\pi} = \delta_1 s$$

Where s the width of the flap. It results from the above that from the opened volume should be removed from the net supply

$$V_1 = V_{1,\pi} - V'_{1,\pi} = (\delta_2 - \delta_1)s = 2\ a\ s$$

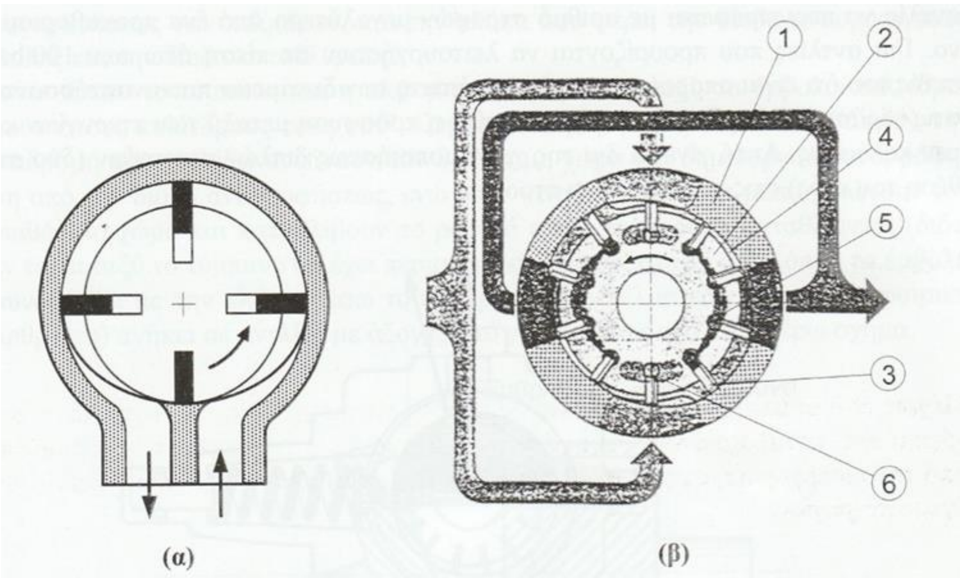
Taking into account that the situation above will be created Z time in rotation, as many as the flaps and that the width of the flaps is b, easily arise:

$$V = 2a(2\pi R - Zs)$$

$$V_p = b2a\left(R - \frac{Zs}{2\pi}\right)$$

6.6.1 Pumps with fixed supply flaps

These pumps have been studied above. As shown at sketch 6.6 below, they are consisted of the hull, on which the bearings of the rotor that has eccentricity to the inner surface of the hull and the flaps are supported. While the rotation of the rotor flaps is tangent on the inner surface of the hull either because of centrifugal force or because of pressure. Special supply can be in the term of 10 to 100 cm3/speed and the operating pressure usually reaches 175 bar.



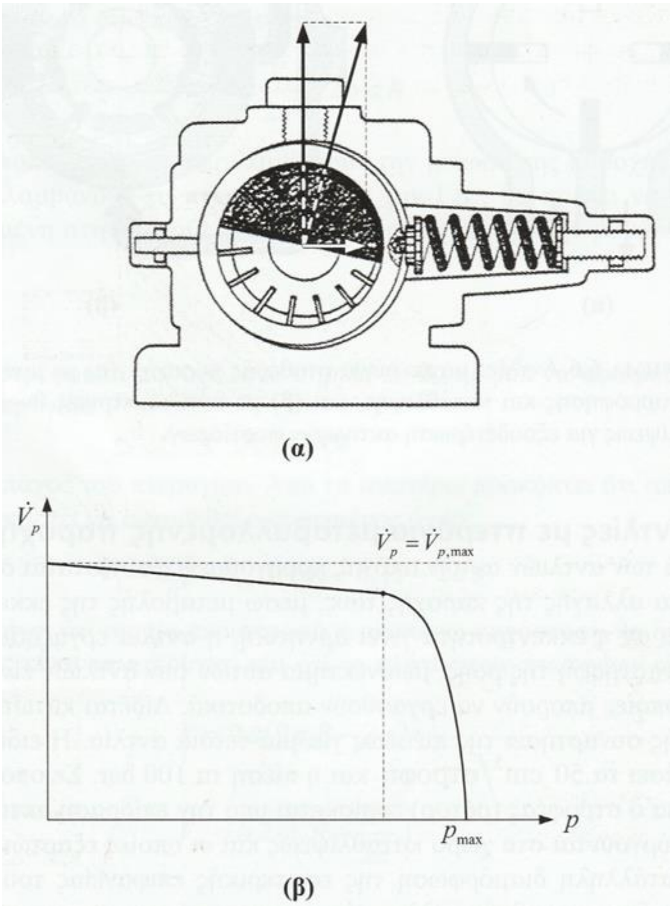
Sketch 6.6 Pumps with blades of constant supply (a) with single suction and depression valves, and (b) with contrary depression valves for the neutralization of radial loads.

6.6.2 Pumps with variable flow flaps

The difference between these pumps and the previous ones is that they have the ability to change their flow by changing their eccentricity. Even if the eccentricity becomes negative, the pump only works if the flow has been reversed. A disadvantage of these pumps are the low pressures in which they can work efficiently. The diagram of the supply in function with pressure for such a pump is given below. Special supply can reach 50 cm3/speed and the pressure 100 bar. In any pump with flaps, the rotor is under the affection of radial forces created in the depression area, and they depend on the pressure. It is possible for many suction and depression spaces to be created with the suitable configuration of the hull's inner surface located in ant diametric positions, in a way that hydraulic balancing of forces occurs.

Depending on the level of pressure for which the pump is going to operate two different ways are used in order the regional adhesion of the inner surface of the hull to take place.

We take advantage of the force that comes from the hydraulic pressure beyond the centrifugal force, for pressures up to 100 bar, which, through the appropriate corrugations is led to the base of the flaps. This mode requires the pump to rotate with a number of speeds higher than the predetermined. It is necessary for pumps that are made to operate at a pressure higher than 100 bar, not only the balancing of the forces that develop in the bearings but also the confrontation of the adhesion between the flaps and the hull. This happens by the use of double flaps (two in place of one) at the sockets of the rotor.

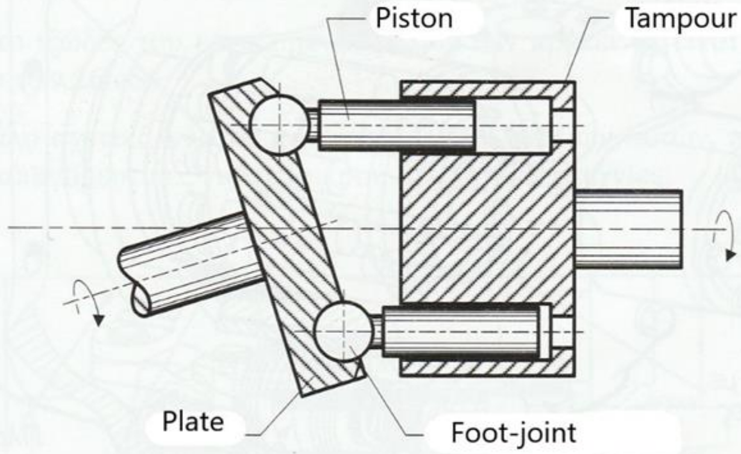


Sketch 6.7 Pump with variable supply blades (a) forces to the rotor, (b) typical special supply-pressure curve.

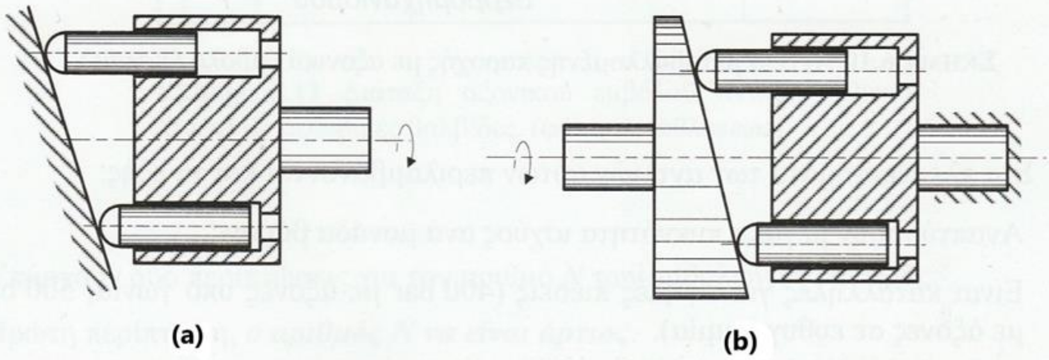
6.7 PUMPS WITH AXIAL BEARINGS

Pumps with pistons create the phenomenon of pumping through the motion of reciprocating pistons inside its cylinders. The mechanism of pumping creation mainly consists of the body of the cylinders (tambour), the pistons with joint and sole, the inclined or not plate (on which the fixed or sliding soles of the pistons perch), the sole retaining springs and the plate that bears the suction and depression passage.

When the tambour and the plate rotate together, the pistons reciprocate in their cylinders all the way. They increase the available space when they erase their route and create suction from the suction passage, but when they move inwards they reduce the available space and depress the fluid through the depression passage (because meantime the tambour will have rotated). That situation where the pistons are connected through the sole with the plate (sole here is a spherical joint) belongs to axial angled pumps, as in the sketch below.

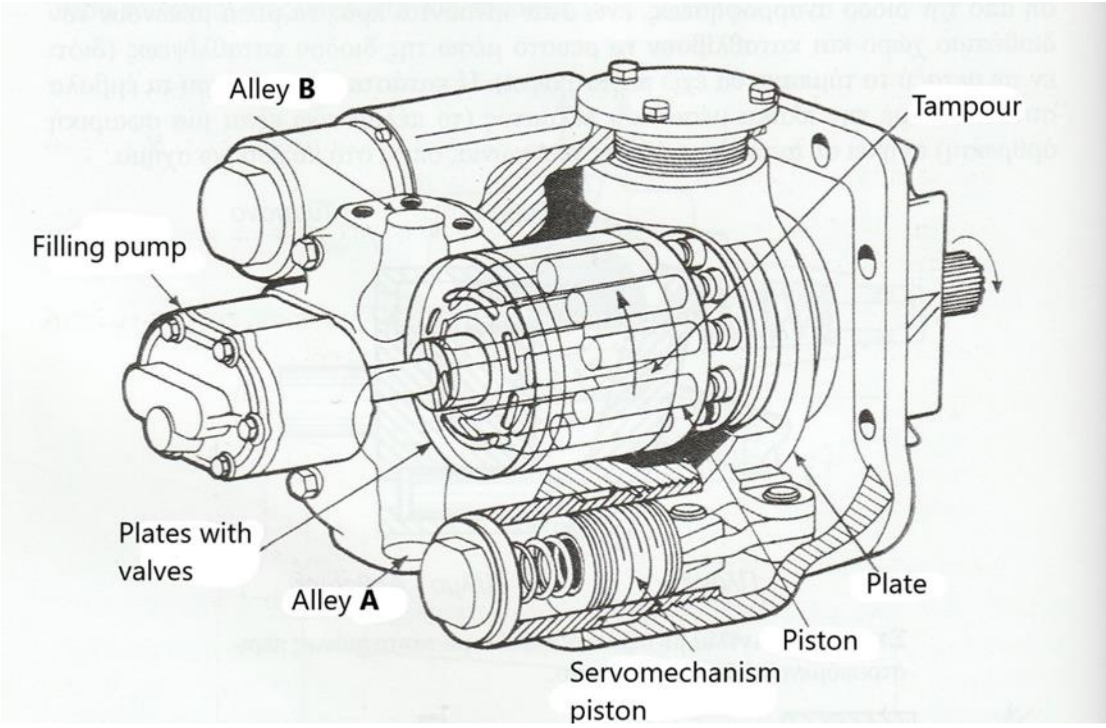


Sketch 6.8 Pump with axial pistons with simultaneously rotating plate nad tampour.



Sketch 6.9 Pumps with axial pistons. (a) rotating tampour and fixed plate. (b) rotating plate and fixed tampour.

Still, there are two more variants of these pumps, where the axes are in line. The pistons here are not linked with the plate but they slide on it. On one variant, the tambour rotates and the plate is stable, sk.6.9 (a), while on the other plate rotates and the tambour is stable, sk.6.9 (b). The continuous contact of the pistons' soles with the plate (especially during suction), is accomplished with springs or special sockets of the soles which do not allow the soles' removal from the plate. Some passages carry fluid at the contacts of the soles with the plate thus achieving the reduction of friction (friction liquids).



Sketch 6.10 Variable supply pump with axial pistons (Vardis).

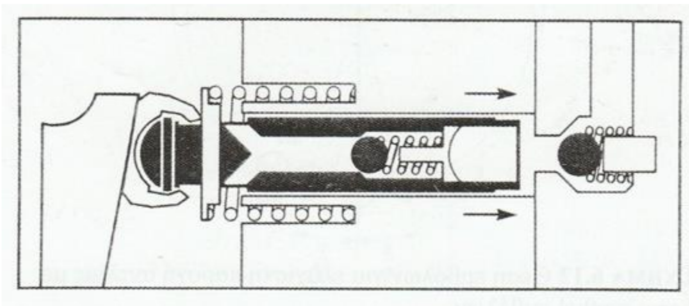
The advantages of these pumps also include the following:

- They develop large power density per weight unit.
- They are suitable for high pressures (400 bar with angled axes, 500 bar with axis in line).
- Pumps with angled axis reach 5000 RPM, while pumps with axis in line a little lower).
- They have slight slow unevenness and silent operation, while they also have slight inertial torque of the moving parts that allows them large accelerations.
- They are of variable special supply with adjustment from zero to the maximum possible rate. The axis angle in axial angled pumps can take

rates up to 45º, while in pumps with axis in line the plate can have inclination up to 25º.

As disadvantages of axial pistons pumps can be considered:

- Their sensitivity to the not well filtered hydraulic liquid. The filter must be of 10 μm for this kind of pumps.
- The fact that the viscosity of the working means must not be small, e.g., smaller than 16 cSt.
- The relatively high construction cost of these pumps, which enables them almost prohibitive for the small crafts.



Sketch 6.11 Axial piston-pump arrangement (Dynex-Rivett) with spherical valves. (Depression phase)

There are two cases for number N of a pump's pistons:

First case, ***number N be an even number***.

The **minimum supply** for that case happens when a piston, let's assume the 1st is on the upper position where the measurement of time starts. Then, due to symmetry, its antidiametric will be at the lower position. The minimum supply of the pump is

$$\dot{V}_{min} = Ld^2\omega\frac{\pi}{8}cot(\pi/N)$$

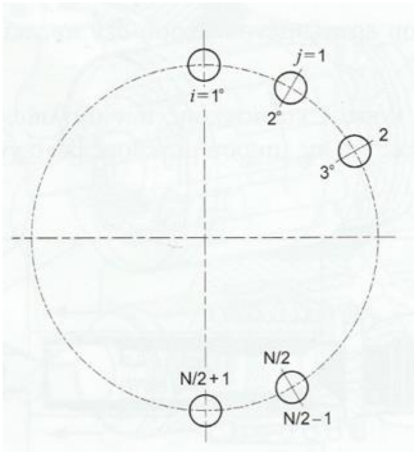
For even number of pistons the **maximum supply** happens when the 1st piston has erased an angle $\theta = \omega t$ which is equal to the half between two consecutive pistons of an angle,$\theta = {}^{\pi}/_{N} = \omega t$,

And if $a = {}^{\pi}/_{N}$, $\theta = {}^{2\pi}/_{N}$ and $k = (N/2) - 1$ results the simplified form of maximum supply,

$$\dot{V}_{max} = Ld^2\omega\frac{\pi/8}{sin(\pi/N)}$$

Supply pulse is defined the difference of the minimum to the maximum supply rate,

$$\delta\dot{V}_a = \dot{V}_{max} - \dot{V}_{min}$$



Sketch 6.12 Piston position for minimum pump supply with even piston number.

After replacement results that

After some simplifications to the form above, the supply pulse for the case of the axial pistons pump results, with even number of pistons,

$$\delta\dot{V}_a = Ld^2\omega\frac{\pi}{8}tan(\pi/2N)$$

Second case, ***the odd number of the pistons***.

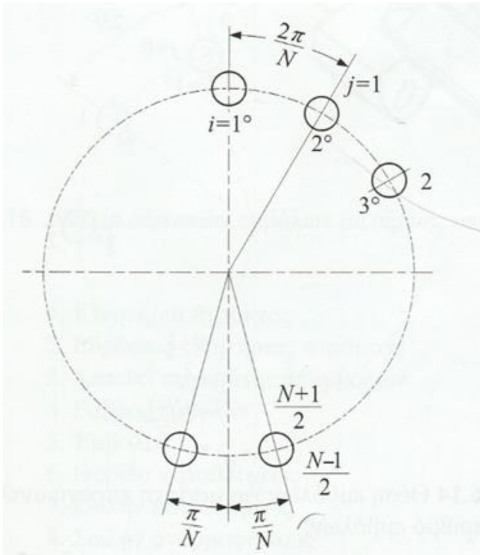
The pump will have minimum supply when the one (e.g., the 1st) piston is to the upper position, so $\omega t = 0$ and the pistons will contribute in a way i that

$$\omega t + [2\pi \cdot (\iota - 1)/N] + \frac{\pi}{N} \leq \pi$$

That is the last piston in the row that will depress and precede the 1st will be the $i = (N+1)/2$, as it results from the solution of the relation above, i.e. they contribute to the pump supply, during the time t, the pistons 2nd, 3rd,…, $(N+1)/2$, of which the corresponding supplies are:

Under these conditions the **minimum supply** of the pump is

$$\dot{V}_{min} = Ld^2\omega\frac{\pi}{8}\sum_{j=1}^{(N-1)/2} sin(j2\pi/N)$$



Sketch 6.13 Piston position for minimum pump supply with odd piston number.

If in the addition relation of the sinus that was given previously is put

$\theta = {}^{2\pi}/_{N}$ And $k = (N-1)/2$ results the simplified form of the minimum supply,

$$\dot{V}_{min} = Ld^2\omega\frac{\pi}{8}\frac{1}{2}cot(\pi/2N)$$

For odd number of pistons N the **maximum** supply will take place when at the space of the depression are found as more as possible pistons, which means that the 1st will form an angle φ_1 with the start of times, such so, as it results from the next figure,

$$\varphi_1 + \frac{2\pi}{N}\left(\frac{N+1}{2} - 1\right) + \varphi_2 = \pi$$

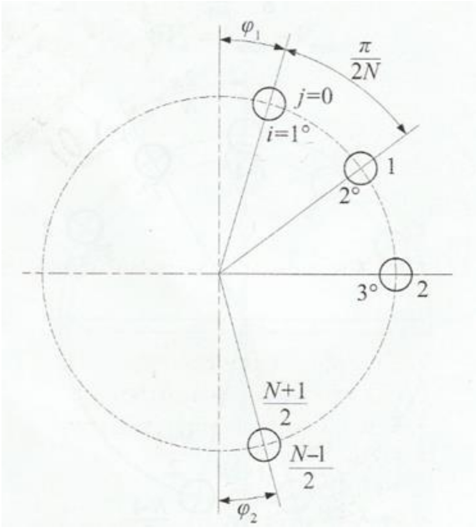
Or

$$\varphi_1 + \varphi_2 = 2\pi$$

Which could be easily result from the previous sketch.

It can be taken on first side, due to symmetry of the depression box

$$\varphi_1 = \varphi_2 = \frac{\pi}{2N}$$



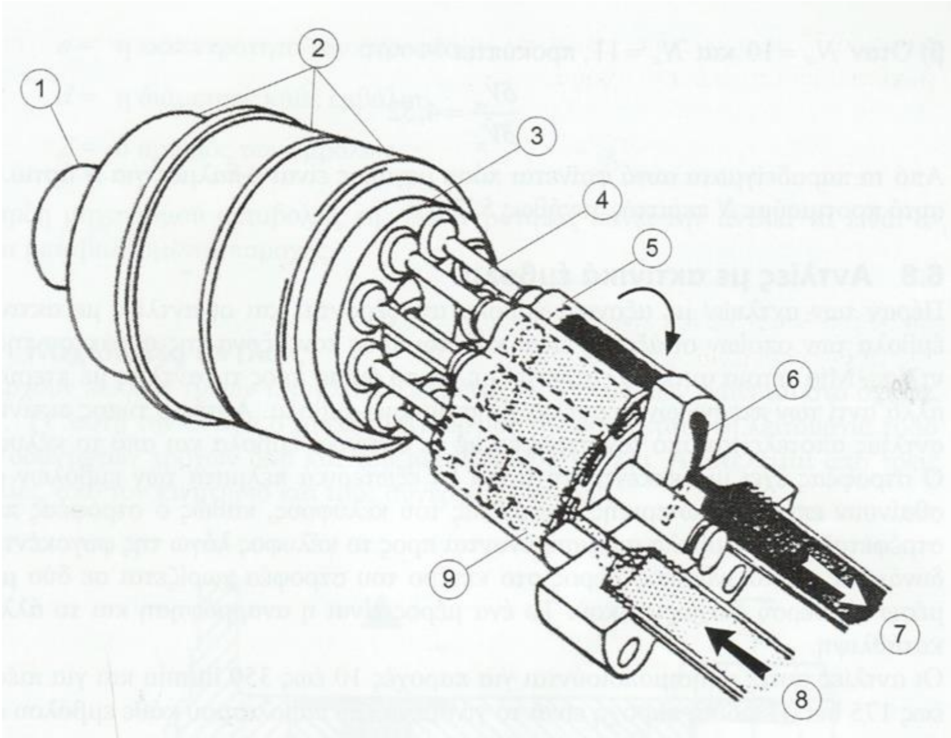
Sketch 6.14 Piston position for maximum pump supply with odd piston number.

If we use the previous relation that was mentioned for the addition of sinus of the type $sin(a + j\theta)$, where here $a = {}^{\pi}/_{N}$, $\theta = {}^{2\pi}/_{N}$ and $k = (N-1)/2$ with $0 \leq j \leq k$ results the simplified form of the maximum supply for odd number of pistons N

$$\dot{V}_{max} = Ld^2\omega \frac{\pi/8}{2sin(\pi/2N)}$$

For that case, N= odd, the supply pulse is

$$\delta\dot{V}_{\pi} = Ld^2\omega\frac{\pi}{8}\frac{1}{2}\ tan(\pi/4N)$$



Sketch 6.15 Axial piston pump with axes at an angle.

1. Drive shaft
2. Drive shaft bearings
3. Piston's restraint lid
4. Piston rod
5. Piston
6. Depression box
7. Depression pipe
8. Suction pipe
9. Cylinder body that carries the bearings (revolver).

Appliance

a) $N_a = 10$ And $N_\pi = 9$. The ratio of supply pulses will be

$$\frac{\delta \dot{V}_a}{\delta \dot{V}_\pi} = 2\frac{tan(\pi/2N_a)}{tan(\pi/4N_a)} = 2\frac{tan(\pi/60)}{tan(\pi/36)} = 3.62$$

b) When $N_a = 10$ and $N_\pi = 11$, results

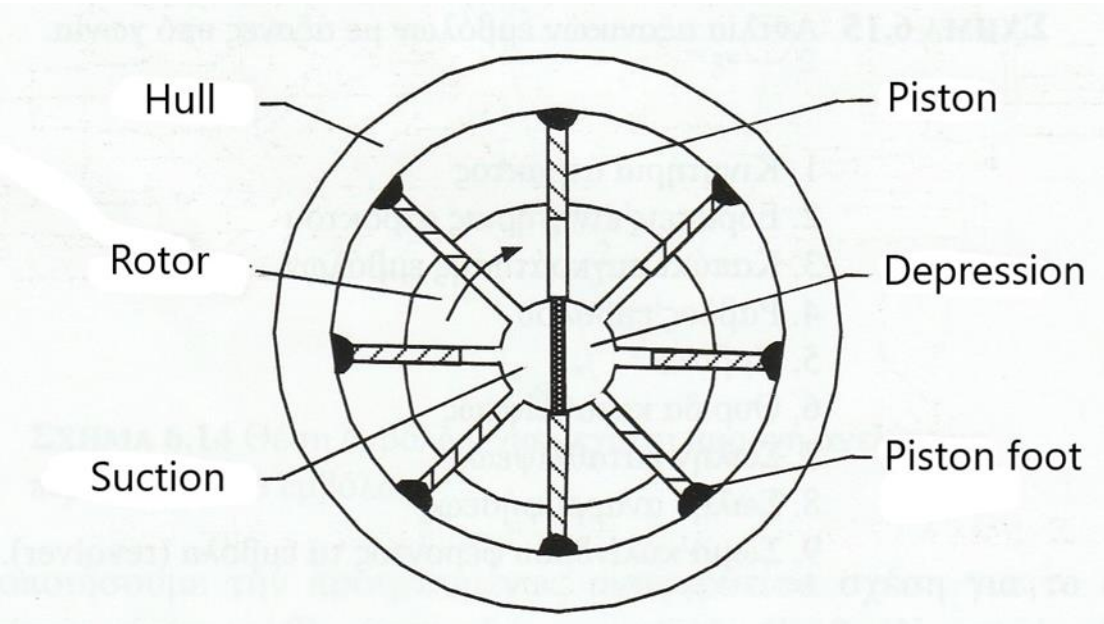
$$\frac{\delta \dot{V}_a}{\delta \dot{V}_\pi} = 4.52$$

From the examples we notice how big the pulse is for an even number N. that's why we choose odd number N, usually 5, 7 or 9.

6.8 PUMPS WITH RADIAL PISTONS

In addition to pumps with axial pistons, there are also pumps with radial pistons whose shafts are perpendicular to the axis of the pump shaft. Such a pump operates in a manner similar to flap pumps but uses pistons instead of flaps. This type of axial pump consists of a rotor with radial pistons and the hull. The rotor has an eccentricity, and the outer soles of the pistons slide on the inner surface of the hull, as the rotor rotates and the pistons due to the centrifugal force are moved towards the hull. The cylindrical area at the center of the rotor is divided by a fixed separator into two parts. One part is the suction, and the other part is the discharge.

These pumps are used for supply rates 10 to 350 lit/min and pressures up to 175 bar. Special supply is the product of the stroke of each piston on



Sketch 6.16 Main parts of axial piston pump.

the number of the pistons and because the way of each piston is equal to twice the eccentricity, we can write

$$q_p = 2\varepsilon \frac{\pi}{4} d^2 Z$$

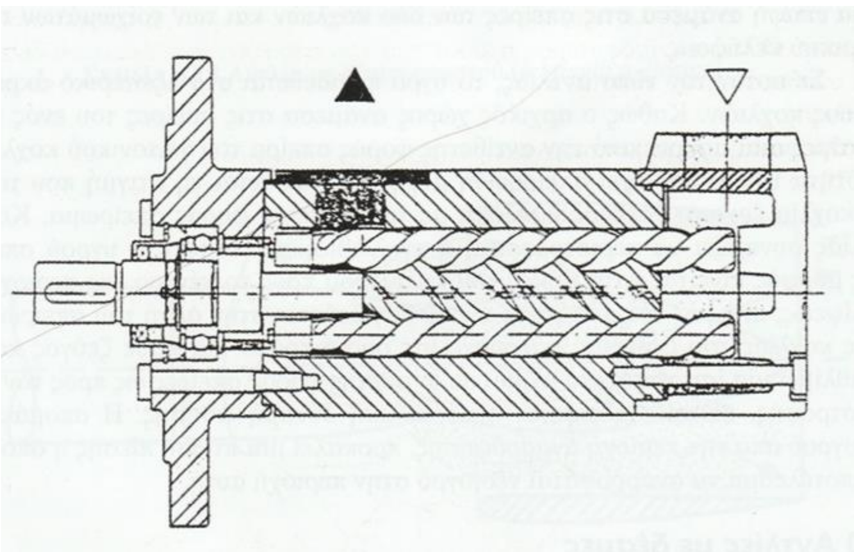
Where

ε = the rotor's eccentricity

d = the diameter of each piston

Z = the number of the pistons

The existence of an eccentricity change mechanism makes the pump a variable supply pump.



Sketch 6.17 Typical bolt pump with three bolts, driving (middle) and cooperating.

The fluid inserts from suction to the gaps between the spirals of central bolt with the cooperating ones and is promoted in parallel with the axis of the bolts to the discharge. The sealing is accomplished with the slight gaps that exist between the bolts and the hull and with the many coupling positions of the bolt coils. They are widely used in cases that large supplies with small pressures are required, such as at lubrication systems or at fluid transfusion systems etc. The length of the bolt must be increased or the step to be reduced for high pressures (in order to exist many coupling positions of the spirals). They appear at almost silent operation and constant special supply, yet they have a low volumetric degree and a relatively high construction cost. As variable kinds of positive bolt-type displacement pumps exist, the main differentiations among them, are recommended to the different number of spirals of both bolts, the step of the bolt and the direction of fluid flow.

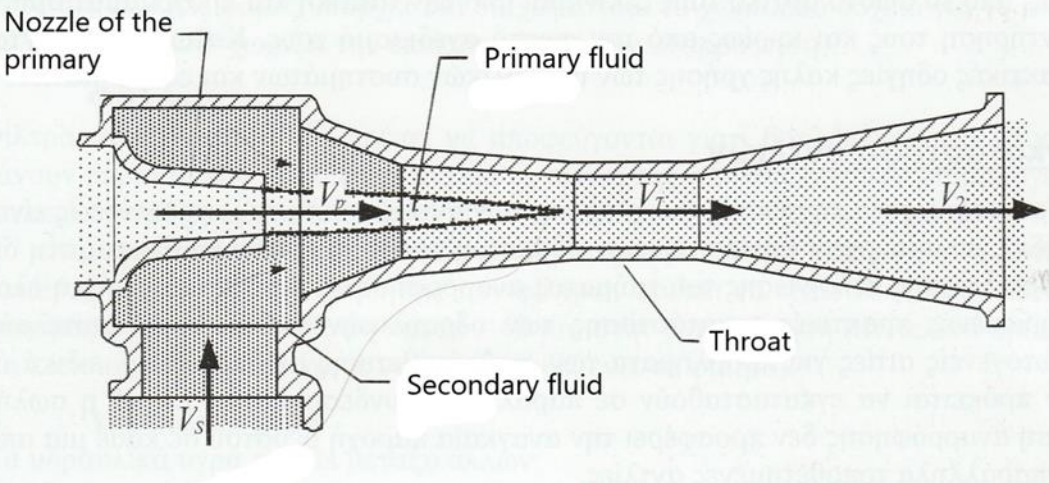
The most widespread bolt pump type consists of two coils that cooperate by being in two parallel instated shafts and couple by sustaining a very little grace between them. The first bolt affords a clockwise coil and the second a counterclockwise coil. One shaft guides the other through a cogwheel motion synchronization stage that manages to move the bolts with a synchronized way and sustain the necessary grace among them. The bolts rotate inside a pair of cylinders that are united and form two overlapping cylindrical cases. Although the dimension graces are small, no contact is made between the coils of the two bolts and the walls of the cylindrical hull.

At that type of pump, the liquid is trapped at the outer edge of each pair of bolts. While the initial space between the spirals of the one bolt is moved away to the opposite direction spiral of the neighboring bolt, an amount of liquid that has a spiral shape, is trapped the moment that the edge of the bolt is coupled again with the opposite direction spiral. As the bolt continuous to rotate, the trapped amount of spiral shaped liquid moves along the cylinder to the center of depression area, while a new amount of liquid is trapped at the beginning of the spiral. Each bolt of the pair operates in the same way and each pair of bolts depresses an equal amount of liquid in an opposite direction coil to the axis of rotation, so the hydraulic pushing force eliminates. The removal of the liquid from the suction area causes a pressure drop, which has as a result the suction of new fluid in this area.

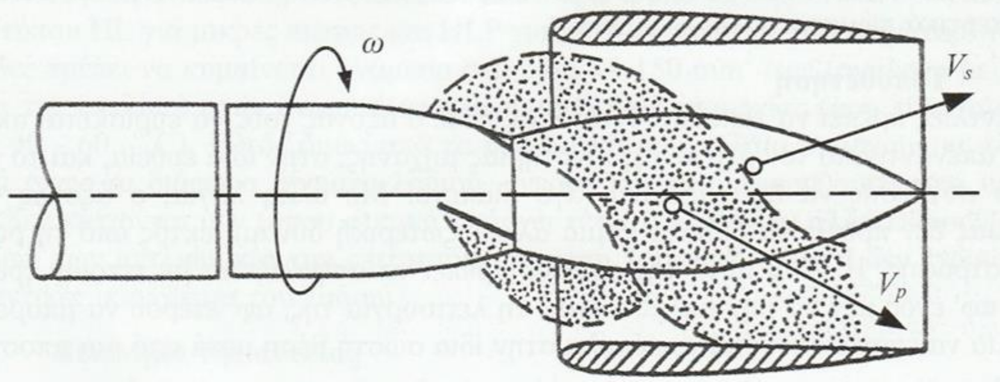
6.10 PUMPS WITH TIES

This type of pumps constitutes the evolution of the common injector, which is a pump without moving components, which uses activated (of high total pressure) fluid (referred as primary fluid) for the activation or motion and transport of another fluid (secondary fluid) as in figure 6.18.

The activated fluid enters the injector through a nozzle, where it creates under pressure, due to which the secondary fluid enters the injector, where the two fluids are mixed and through the convergent-divergent duct the process of activating or pumping the secondary fluid is completed. The energy transfer from the primary to the secondary fluid is only achieved through the friction forces and by the mixture of the primary p with the secondary fluid s and therefore the efficiency degree is too low. In the sketch of the injector the velocities V at the various positions (von Karman) are shown.



Sketch 6.18 Pump with fluid budle (jet pump or ejector)



Sketch 3.19 Pump with rotating fluid budles (rotary-jet pump or Foa's pseudobladed jet pump)

Rotary jet pumps launch the primary fluid in the shape of more beams through gaps obliquely arranged on the rotating pipe, creating in that way blades (pseudo blades), which activate the secondary fluid through moving surfaces (interfaces) and by the appliance of high tendency very efficiently. Sketch 6.19 of this pump shows the rotating with angular speed ω bearer of the nozzles of the primary fluid and velocities V_p of the primary and $V_s$of the secondary to the immovable reporting system.

6.11 INSTALLATION AND MAINTENANCE OF HYDRAULIC SYSTEMS

The right operation and the long life of industrial hydraulic systems of high pressure depends a lot on their correct installation, their right initial start, their regular and scheduled maintenance and mainly their correct design. Practical instructions for good use of the hydraulic systems and components are given below.

Pump installation

The requirements for the proper installation of positive displacement pumps are the same as these for the centrifugal. Special attention must be given to the configuration of the piping of the suction department. Wrong or incomplete practices of installation of hydraulic systems constitute primary causes for the problems of positive displacement pumps, especially when it comes to their installation in parallel connection. The suction piping often does not offer the necessary fluid supply in each of the parallel assembled pumps.

1. Cleaning

Pumps must be checked externally whether they have any damage during their transportation and get cleaned before they are installed to the hydraulic assembly, especially when the input and the output are not shut with protective caps.

2. Setting

Pumps must be set in a way that their axis be located precisely across the axis of the drive machine, on the same line and the connection coupler to have axially a slight gap. In other words, the axis of the pump must not take any other external force apart from the rotation torque. This position of the pump must be secured in such a way that on the one hand it does not change during its operation, on the other hand the pump can be easily repositioned at the same position after a log out.

The suction pipe must be sealed from the suction open to the other edge and quite sunken into the hydraulic liquid so that it cannot draw air with the under pressure. It must also have a quite large diameter, so that the speed into the pump to be low (0.5-1.0 m/sec) in order for the suction under pressure to remain at permissible limits. Finally, curves and long lengths should be avoided and its edge not to abstain less that the triple of its diameter from the bottom of the container (so it does not pull dirt) and be obliquely cut (45^o).

Leak pipe (wherever it exists) must be quite large and be freely driven to the container so that the pressure does not rise in the hull of the pump beyond the permitted limits because this may destroy the seal of its axis. Pipes that contact the container must be submerged in it, at least 100mm, so that they neither create foam (of the returning) nor draw air (of the suction) and they are quite far apart, so the hot usually returning oil does not suck before it cools (there is sometimes a separating wall for that purpose, which separates the returning from the suction).

3. Filters

Filters must be avoided in suction because they clog easily, and they cause more damage than good to the pump due to cavitation created from the increase of the under pressure in suction. It is preferable for that reason to use filters in the returning or depression. Their filtration ability must be 10-20 μm in returning and depression and 25 - 40 μm in suction (if, finally, the filter must be placed there for any special reasons).

4. Hydraulic liquids

Hydraulic liquids must among other things:

A. Be able to carry hydraulic power and bear high pressures.
B. Lubricate the pump and the moving parts of the other hydraulic elements.

These requirements fulfil in the best way the hydraulic mineral oils of type HL for small pressures and HPL for high pressures. Their kinematic viscosity must fluctuate among the limits 15-150 mm2/sec (depending on the type of the pump) at the expected operating temperature (which is usually 40 - 60ºC). Nevertheless, apart from hydraulic mineral oils there are also other liquids such as the variable oil-water mixtures and synthetic liquids that do not burn (wherever this kind of danger exists) but eliminate pumps' life and the permitted operating pressure, because they don't have the lubrication skills of the oil.

5. Pump start

Before a hydraulic pump starts for the first time we should first pay attention if there is the right oil at the container and to the right voltage supply (provided that the pump is powered by an electric motor). Pumps that are lubricated by their leaks is necessary to be filled with oil before start, after unscrewing the leak pipe, because otherwise they will start without lubricating oil and they may get damaged.

At first start we press the START button and then immediately the STOP button watching (from the engine's fan or the coupler) if the engine rotates properly, according to the direction of the pump rotation (there is usually an arrow on it). If it is not rotating correctly, we change two of the three phases of the electric motor so that the direction of rotation will change. We can now set the pump to operate. In most cases, we have free flow of the oil from the container, in the position of calmness of the valves of the hydraulic system and so the bleeding off the pump is made automatically. Where this does not happen and there is not a bleeding off pump we must do the bleeding off by unscrewing for a while the pressure pipe to a place and then bolt it again. If we do not notice any unusual noise, then everything is all right. All these are done with the security valve (for safety reasons and for a better bleeding off) open and then we adjust it to the maximum permissible pressure.

6. Maintenance

Periodically, we check the noise of the pump and tighten the piping if necessary. Depending on the manufacturer's instructions and at any case that we will understand a noise and temperature increase or find out pressure pump supply decrease we should definitely check it and if we find out any damage to repair it before its condition gets worse.

7. Pumps faults

The most usual faults of the pumps are seen either from the increase of their usual noise or the temperature increase of the oil, or the pressure drop or/and of the supply.

Unusually big noise

If this is not due to wrong rotating direction or to too many speeds, then it comes either from air suction (from the pump input or the seal of its axis) or from obstruction of suction by a cause, or from a mechanic part that will have great wear. The noise in the first two cases is a consequence of cavitation and characteristically loud.

Very high oil temperature

It comes from the increased inner leaks due to wear to the moving parts or from mechanical friction of pump components and engines.

Supply drop

It comes from inner leaks due to wear.

Pressure drop

It is a result of advanced leakage. The pressure and supply drop as also the temperature increase may be of course due to other causes that are found in the rest hydraulic components (mainly security valves or non-return valves that do not close properly or worn valves with a bolt. Regular research from the beginning to the end and very much practice is required to specify the cause of the fault.

The real hydraulic liquids are neither incompressible nor without friction. This determines significantly the dynamic behavior of the hydraulic systems. We must not forget the pressure losses inside the piping. The compressibility of the hydraulic liquids together with the elasticity of the cylinders and the piping as the contained air is at certain cases calculable and creates disturbing effects, especially when pressures are high and the under-pressure volume is large. This compressibility is perceived when e.g., the pressure drops abruptly in a big cylinder just before reversing its motion or in an abrupt stop. Frictions have consequently, as we already said, pressure drop (losses). Both of these properties, compressibility and frictions, can many times, under certain conditions, affect strike and noise damping at hydraulic systems positively.

6.12 DESIGN AND CALCULATION OF HYDRAULIC SYSTEMS

Here a reference to the design and calculation of the hydraulic systems is done. It is necessary for all the required operations of the system to be determined, and the best technique and also a financial solution to be elected for the appropriate design of a hydraulic system.

It is easy for a hydraulic system that satisfies the operation requirements to be designed. However, if the right components are not selected, or if complicated and over-dimensioned components are selected, it is possible to arise an expensive system that can satisfy the requirements, but not proceed with the construction due to the increased cost.

It is always important for a hydraulic system to be correctly designed, with the appropriate components, which has the best possible cost for the specific operating requirements and the maximum possible economy at power consumption.

The following characteristic parts are mainly reported as the requirements of a hydraulic system:

Boost and traction force in hydraulic systems.

Linear speed in hydraulic pistons.

Torsional torque at hydraulic engines.

Number of speeds in hydraulic engines.

Time and operation circle diagram for each hydraulic piston and engine.

Requirement for minimum dimensions of the hydraulic engine or cylinder, or of other components (that is higher operating pressures).

Use of electric engine or internal combustion engine.

Any restriction on the given power to the pump.

Distance between the various components and especially from the pump (it is related to the length of the piping and pressure losses).

Way of valve handling (manually, mechanically, hydraulically pneumatic, electrically).

Climatic conditions in which the machine will operate.

Use of special hydraulic fluids (non-flammable etc.).

Special protection requirements (e.g., operation in an environment with lots of floating dust particles that is possible to enter the fluid and cause rapid wear).

Area for the system's replacement.

Additional adjustments and marks.

Requirements for connection with central electric or electronic control system.

Requirements for operation with specified voltage, current and frequency of electricity.

Requirements for increased reliability and minimization of maintenance.

Requirements for the use of specific components.

Cost limitations.

Generally, always a try is made to use standard components. This happens for the minimization of cost, the easy procurement of the components from the construction companies and the simplicity and standardization of the hydraulic system's construction.

When the operations requirements are fully known, the hydraulic circuit is initially designed based on the standard hydraulic symbols and the connections with piping among them for fluid supply. Subsequently, the number of components is calculated, and an effort is made for the best solution on the design. For all the elements that are not standard (such as hydraulic cylinders if they are going to be constructed and are not standard) is required a control in resistance to be done. The supply and pressure of the fluid is determined at each point of the system.

Great attention is needed for the dynamic behavior of the whole system, so as to avoid over-pressures, noise and unstable oscillating behavior, especially in case that the accelerations or speeds or mazes that are going to be displaced are great. Thus, in some cases the designing and the calculations with the help of modeling on a computer are necessary.

6.13 SELECTION OF STANDARD COMPONENTS

The hydraulic system and all calculations must be done in order the selection of the components to take place. After the selection of the components, because these are standard it is very likely for some dimensions to change so it necessary for the final calculations of the system to be done and the final outputs to be calculated (displacements or rotations) according to the standard components. The components according to the operation requirements of the specific hydraulic system are selected from the manufacturers' lists. As it has been mentioned, standard-popular components must be selected as much as possible.

The next selection elements are important for the **hydraulic pump**:

Type of pump (gear, piston, blades etc.).

Fixed or variable supply and type of adjustor (only for variable supply). Open or closed circuit.

Size (cm^3/rev.). Maximum operation speed (rpm).

Maximum pressure (bar). Direction of rotation. Hydraulic operation liquid.

Type of axis. Type of adjustment and bearing. Pump combination.

Spirals or flanges for oil fittings. Special additive elements.

If a **hydraulic engine** is going to be selected, its selection elements are:

Type of engine (gear, piston, etc.).

Fixed or variable capacity and type of adjustor (only for variable capacity).

Size (cm^3/rev.). Maximum operation speed (rpm).

Maximum pressure (bar). Direction of rotation. Hydraulic operation liquid.

Type of axis. Type of adjustment and bearing. Pump combination.

Spirals or flanges for oil fittings. Special additive elements.

Selection elements for a **hydraulic piston-cylinder** are mentioned below:

Type of piston or cylinder (simple, double energy, special construction etc.).

Maximum pressure (bar). Types of bearing.

Diameter of piston (mm). Rod diameter (mm).

Path (mm). Joints (if needed). Construction material.

Surface coating. Hydraulic operation liquid.

Specifications. Measurement systems. Special additive elements.

The selection elements for **non-returnable** valve are:

Type of non-returnable valve (internal, with spirals, with hydraulic command).

Size – is related to the oil supply.

Maximum pressure (bar). Pressure drop Δp (bar). Hydraulic operation liquid.

Type of adjustment and bearing. Special additive elements.

The following elements are the most significant for the selection of a **standard flow direction valve**:

Type of flow direction valve (manual, with hydraulic command, with pneumatic command, electromagnetic).

Number of oil way and operating positions of the valve.

Size – is related with the oil supply.

Maximum pressure (bar). Pressure drop Δp (bar).

Type of valve 'latch'. Hydraulic operation liquid.

Type of adjustment and bearing. Time of execution of commands.

Electric connections (for electromagnetic valve).

Protection – standards and special additive elements.

Subsequently the selection elements of **pressure regulator valve** or **security valve** are mentioned:

Type of security valve (direct or indirect operation, security valve or pressure reducer, operation, purpose).

Size – is related to the oil supply.

Maximum pressure (bar). Pressure regulation limits.

Pressure drop Δp (bar). Manner and type of regulation.

Hydraulic operation liquid. Type of adjustment and bearing.

Protection – standards. Special additive elements.

The selection elements for the **flow regulation valve** or **shut off valve** is:

Type of shut off valve (shut off valve or flow regulator, with non-returnable valve or not, operation, purpose).

Size – is related with the oil supply.

Maximum pressure (bar). Supply regulation limits.

Pressure drop Δp (bar). Manner and type of regulation.

Hydraulic operation liquid. Type of regulation and bearing.

Protection – standards. Special additive elements.

The next elements must be noticed for the correct element's selection of the **analogue valve**:

Type of analogue valve (with or without pilot, feedback, purpose, operation).

Number of oil ways and operation positions of the valve. Maximum pressure (bar).

Size – is related to the oil supply.

Pressure drop Δp (bar) and correct and acceptable operation limits (calculations). Dynamic elements.

Electronic cards. 'Latch' type valve.

Hydraulic operation liquid. Type of regulation and bearing.

Time of execution of commands. Electric connections.

Protection – standards. Special additive elements.

The selection elements of **servo valve** as a special component with increased cost:

Type of servo valve (with or without pilot, type of feedback, purpose, operation).

Number of oil ways and operation positions of the valve. Maximum pressure (bar).

Size – is related to the oil supply.

Pressure drop Δp (bar) and correct and acceptable operation limits (calculations).

Dynamic elements. Electronic cards. 'Latch' valve type.

Hydraulic operation liquid. Type of regulation and adjustment.

Time of execution of commands. Electric connections.

Protection – standards. Special additional elements.

The following elements must be selected for the **accumulator**:

Type of accumulator (with gas, with weight, with spring).

Size – is related to the amount of oil that is accumulated.

Maximum pressure (bar). Hydraulic operation liquid.

Attachments for oil supply. Initial gas pressure (if it is with gas).

Auxiliary component (security valve, switch, manometer).

Type of regulation and bearing. Special additive elements.

The next selection elements are mentioned for the **hydraulic fluid filter**:

Filter type (returning, suction, pressure line, special filters).

Type of filter element.

Size – is related to the oil supply and temperature (calculations). Ability of filtering in mm.

Maximum pressure (bar). Hydraulic operation liquid.

Attachments for oil passage. Type of cleanse indication.

Type of regulation and bearing. Special additional elements.

Some selection elements are also mentioned for the **auxiliary components** of the hydraulic systems:

Electric engine: power, operation speed.

Elastic joint (coupler): power, operation speed, inner diameter for engine-pump axis.

Pressure switch: maximum pressure, regulation limits, attachments.

Manometer: maximum pressure, attachments.

Oil cooler: water-cooled, air-cooled, maximum oil pressure-supply, water or air supply, cooling power, attachments, bearing.

Oil level indicator: indication length, regulation.

Temperature control: regulation limits, attachments.

Ventilator: type, size (is related to the oil bottle size – air supply), air filtering, bearing.

Oil container: type, size (calculation), dimensions, electric engine bearing.

CHAPTER 9: EXPERTISE AND DIAGNOSIS IN MAINTENANCE

This chapter follows a step-by-step approach of machine maintenance training as it is used in industrial training, for the purpose of the efficient transfer of expertise to the educating major or technician related to the machine fault diagnosis.

9.1 INTRODUCTION

Every person who is interested in the collection of measurements and mechanical installation, will learn the process of fault diagnosis and wear, and damages recovery in a precise and unique way, as well as how we examine machines, which machines are examined, where sensors are placed, what kind of sensors are used, how we collect the measurement results, how the measurement results are processed and analyzed, and how we diagnose the fault or the faults of machines, etc. The experts of this subject can verify their knowledge and feel better.

Almost all industries use more than one maintenance methods. The variable measurement methods provide data that lead to a better view of a machine's condition. The maintenance cost affects financially an industry. Cumulatively, the final cost is also due to the cost of the oversized STOCK of spare parts, long hours of maintenance technicians, profit losses from the production halt and secondary faults due to an emergency stop of the machines. At appropriate maintenance results the reduced energy consumption and the good quality of the production can also be added. Every industry has set its own maintenance methods and success moderation. The purpose is to get in line with the guidelines committing our best for goals' achievement. With continuous production machines it is important to keep the machine at good operating condition. Any mistake causes a huge production cost and a big material loss.

For example, when a ship has a specific timeline for catches through the year and gets a damage that we cannot deal immediately with, then the financial loss leads to disaster. Shutdown (blackout) leads to huge financial losses and also criminal liability due to damages to citizens' property, at energy production industry. All these are general and interesting, but each one must at the right time learn well the machines for which they will be responsible. If you intend to detect a machine malfunction in a time manner and make changes to increase reliability, then you should learn really well the reasons why machines have faults, which are responsible for the machines' failure.

The beginning of a machine's failure can be detected even in the designer's or scholar's office, while the failure is completed with poor maintenance and due to operation conditions. Design, construction and installation manner and a machine's inspections determine its life lasting. Obviously your speech for all the previous steps will not pass. Yet, an appropriate knowledge of the maintenance rules and the reasons that cause faults will lead to good results.

Here our aim is not to learn the way of organizing a maintenance department, because this also depends on the type of machines and the priorities that will exist.

Our purpose is to know the reasons that cause faults and have the ability to recognize the chances of improving a machine's reliability, decrease the maintenance cost, decrease power consumption and improve the quality of the manufactured products. The knowledge of the rules for fair and effective maintenance does not lead to the results above if all employees have not been initiated correctly into industry. Everyone must understand the benefits of precision maintenance, from the electrician that makes the motors' wrapping to the chief financial officer who signs the checks.

Maintainer's work would be easy if the only thing he had to do was to buy a device for measurements of shocks or/and vibrations collection and a computer and software. Then, all of our problems would have been solved. And of course, all of these would be easier if the machines did not fail. However, things are not that way because reality says otherwise. Machines need maintenance in order not to break. Everyone must understand that proper maintenance costs and must be prepared by specialized technicians with tools and organs. If each one that gets involved to a Condition Monitoring program has no idea and does not believe in the aims of the program, then the success of the program will be limited.

Also, if the rest of the staff in various activities or in manufacturing, or in the sales department, or in the investigation and construction department etc. does not understand the aims of the program, then success will be limited again. They must know the reasons for which proper maintenance is done. Our purpose must be the proper maintenance, but also to inform the rest about it. Failure is guaranteed due to the existence of competition among the different departments (e.g., department of maintenance and manufacturing).

In industries often the following happens: a failure in a machine is detected, the working series is programmed and the time to open the broken machine comes. The electric department technicians wish the mistake to be given to the machinist department, while the machinists wish the mistake to be due to the electricians. These situations create tensions and restrict the success of any program. The solution results through the maintainers' training. Subsequently, we should explain to the others what exactly they must not do and what exactly we expect and in which way they can contribute to the program's success. Only then we will expect success in everything else.

<u>The purpose of training at maintenance of rotating machines</u>

The purpose of this training is to learn ways of first of all recognizing the situation in which it is and second the conditions under which a rotating machine operates. The method must be simple, reliable, and financially advantageous and based on proven technology, in order to be successful.

It is proven that the shock waves and vibrations-shocks monitoring of the rotating machines are possible to detect some warning signals very early depending on the damage that causes them, such as the destruction degree of a rolling bearing, the unbalancing degree of cogwheels and pulleys, the lack of alignment of the shafts, the lubrication conditions, etc.

There are mainly three methods for the study of the emitting signals:

SPM method for impulses measurement, VIB method of vibration intensity measurement and EVAM method which deals with emitting signals analysis by exploiting the potentials of computers and with the use of the appropriate environment.

SPM (Shock Pulse Method) was first used by SPM Company and is referred to the pulse of the wave that is developed after the collision of a moving body on a clog. By applying this measurement of the impulses on the rolling bearings we can recognize most of the cases of fault existence on damaged bearings or on bearings with defective lubrication.

VIB (Vibration) method of oscillation intense measurement (vibrations, shocks) is referred to the comparison of measured shocks tension with the limits that are allowed by the international standards. This method is a low cost, quick and practical way to check a machine’s general condition and detect cases of unbalance, poor alignment, and loose joints.

EVAM (Evaluated Vibration Analysis Method), which deals with the analysis and evaluation of oscillations, uses Data Logger for the collection of waves of the oscillations, but also the rest measuring elements from the sensors’ shooting point and through Software proceeds to the signal processing and analysis. Subsequently, the maintenance technicians proceed to a justification of the fault whose deal is now simple, with the help of EVAM.

Why do we measure and analyze the emitting signals

When a machine shows damage, which will finally drive it to shut with the consequence of probably secondary damages, it usually declares it by emitting noises and vibrations that characterize the specific damage. If we measure the vibration intensity and search the way of vibration and mainly the way of tension change, we can detect the problem very early and act suitably and predictably. Measurements testify the nature of the problem (that is if the problem is due to unbalance or a bearing fault etc.) and the danger of the fault. This means that the measurements can give us enough information, with which we can schedule our activities related to the maintenance of the facilities. Shocks measurements also give us information, which help us to investigate the reasons that caused the problem.

Because a machine is built to last much longer than usual, due to a fault, if we identify the fault and its cause in time and utilize this information, then we can change the way the machine works, as well as the way of installation and maintenance, or even to contribute positively to the design of newer and better machines, which will be more reliable and will require less maintenance. Therefore, the monitoring of the signals emitted by the machines leads to a reduction of maintenance costs and improves the production conditions or the quality of the products. Each company follows different maintenance philosophies of its rotating machines and implements different monitoring and maintenance practices. This is both correct and necessary as monitoring a single component (such as vibration) does not provide answers to all problems. It is important to know the pluses and minuses of every available solution because the combination of different methods of measuring and evaluating the symptoms gives a clearer picture of the condition of the machine.

The purpose of the maintenance technician is:

- To locate the problem long before the failure occurs.
- Make improvements and suggestions.
- To increase the reliability of the machines of which he is responsible.
- To understand the main causes of the onset of failures and problems.

9.2 PRACTICES AND TYPES OF MAINTENANCE

In many industries, in the past but also today, the philosophy was just to keep the facilities operating. If a fault existed, the repair or change of the spare part with a new one was done and there was no margin for improvement of the machine's reliability or for fault prediction and the interest was focused mainly on how quick the machine or the system would get into operation. Such a company considered the maintenance department as a great cause of expenses or loss of money.

Lately this philosophy has changed. Companies started to recognize that it is worth investing time and money to change the maintenance practices with the aim of reliability improvement of their facilities through the maintenance improvement process, which is based exactly on this machine reliability.

Repairing maintenance

According to this, a machine works "until it is broken" and it is essentially repaired or corrected after its disaster or shut due to a fault. So, a machine is left to operate without protection until it stops, that is until it breaks down and continuously we correct the damage by reacting to this. This type of maintenance is recommended with various names, such as repairing or corrective maintenance and breakdown or reactive maintenance.

This practice usually leads to a very high cost and high maintenance expenses. It is also accompanied by secondary lesions of the facility (e.g., the axis may be destroyed when the bearing breaks down), by manufacturing reduction or shut down, by expenses increase due to spare parts STOCK increase, by increase of the maintenance staff's long hours for fault emergency repair etc., which increase cost.

As a component or a machine's element failure can occur at any time, therefore the production and security are affected, so the facilities operation control gets lost. Even if an experienced technician detects an upcoming fault and avoids the worst situation (that is the secondary damage), he will not avoid the production loss during the fault repair, because repairing maintenance is not scheduled.

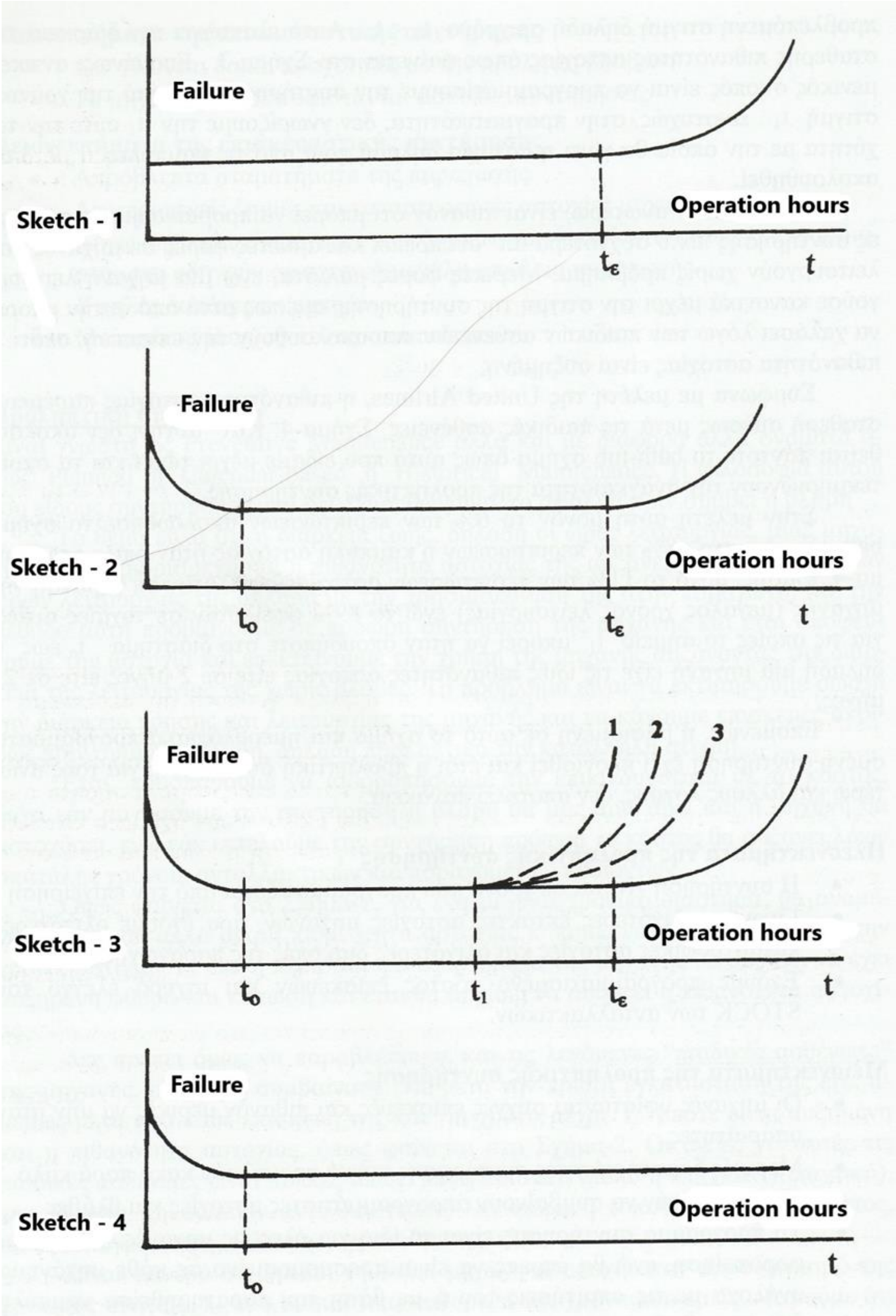
Although this practice was usual until the decade of 1950, even today it exists in some cases, as in some not important for the production machines, for mainly two reasons: firstly, the expenses of monitoring and controlling of a machine may not be justified by the machine's operational results (e.g. if it is out of the production line, or if it does not affect the security) and secondly, may be considered more advantageous for a machine, even if it was monitored until now, to be left from now on to operate until it breaks down.

Advantages of repairing maintenance

- Expenses related to the fault prevention are not made
- Machines do not undergo needless maintenance

Disadvantages of repairing maintenance

- Unpredicted shuts of the production
- Secondary damages and damaging machine failures
- Out of operation time intervals lead to relatively high production loss due to the complication of the faults
- High repair expenses
- Reduced facilities check



Preventive maintenance

It is known as preventive maintenance, but also with other names such as: planned or calendar-based or historical maintenance. The philosophy of that type of maintenance is that “we correct the fault before the damage occurs”.

The point is that the lifespan, which means the operation hours of a machine, is limited and the possibility of failure increases by the increase of its operating hours, that is, by its use in production. By this method we proceed to early maintenance of the machine and so failure is avoided, and the use of the machine is extended by increasing its operating duration without faults. The issue is to estimate correctly the duration of use and operation of the machine and repair it just before the damage occurs, which means preventively at the right time.

So, we try to balance the risk with the cost, because if we delay maintenance the fault will prevent us and the machine will fail but if maintenance is done early, the cost will be increased due to waste of time, spare parts and produced commodities.

As sketch-1 shows, we would expect constant but small failure possibility of a new machine, for a lone time. During the moment t_ε one or more machine components will begin to get increased wear and fatigue, and so the possibility of failure will begin to increase.

Nevertheless, we must not pass by the so-called “childish diseases” of the machine, which occur either after its first installation or immediately after each repair and for time until t_0 when the possibility of failure is increased, as sketch-2 shows. The causes for these childish diseases are many like: poor lubrication (too much or minimal lubricant), placement of incorrect spare parts, wrong placement of the component, poor alignment, lack of balancing etc.

It is obvious that great attention must be given to the definition of moment t_ε which essentially determines the time estimated for the dangerous wear to begin. But this wear usually starts earlier from the predicted moment that is at the moment $t_1 < t_\varepsilon$. This decreases the duration of constant failure possibility as shown in sketch 3. Subsequently, our objective purpose is to schedule the maintenance before moment t_1. Unfortunately, actually we are aware of neither t_1 nor the speed in which the failure will occur, which means which curve, 1, 2 or three, is going to be followed.

Subsequently to the above, it is possible that we may proceed to maintenance actions more often than needed, and many times with machines that operate without problems. Sometimes, even while a machine operated properly until the moment of maintenance, right after that, it may break down due to the childish diseases that follow the repair, so the possibility of failure is increased.

According to United Airlines study, the chance of failure remained constant immediately after the childish diseases, sketch-4. According to that, bath-tub shape, as we have seen until now, which documents the necessity of preventive maintenance, is not always followed.

Only the 6% of the cases followed the bath-tub shape in that study while at the 68% of the cases the failure curve was as in sketch-4. Also, only the 11% of the failure cases was due to aging of the machine (large operation time), while the 89% was due to accidental causes, for which point t_1 can be anywhere in the interval $t_0$to t_ε ,which means that the machine had the same possibilities of failure either in 2 or in 22 months.

Therefore, the based on that shape and calendar scheduled maintenance has been toppled, and thus for the above and other reasons, preventive maintenance is not a nostrum.

Advantages of preventive maintenance

- Maintenance is done at the time scheduled by the enterprise.
- Less emergency failures of the machines are made, so we have less destroying failures and less manufacturing cuts
- We have a scheduled repairing cost and a strong control of the spare parts STOCK

Disadvantages of preventive maintenance

- Machines undergo frequent repairs and maybe some of them are unnecessary
- Some maintenance activities often cause bad than good
- Unscheduled failures and faults continue to occur
- Maintenance schedule is the same for all machines, without difference, while it should have been adapted to each machine depending on its requirements or based on its noticed lifespan curve.

Predictive maintenance

Good news for maintenance technicians is that the machines usually emit signals before they fail. Each signal can refer to the change of the vibrations or shocks, the temperature and the emitting sound increase, the decrease of the efficiency degree of the machine, the existence of metal particles, the change of the engine's electric consumption, or to various other changes. The study and utilization of these signals lead to the philosophy of predictive maintenance, which has also other names and the most usual are Condition-Based maintenance or detective maintenance or prognostic maintenance because we already know what is broken and what is going to be repaired. The philosophy used is "if it is not ready to break don't fix it."

If we achieve the machine's monitoring, the signals recorded, and subsequently schedule their repair and maintenance before the moment of a high risk (or the chance of failure), then we will be able to have machines that deplete their estimated lifespan, thus reduce cost. That means to predict the failure and act at the right time. According to this the maintenance expenses are decreased, and because the machines do not fail unexpectedly and there are no emergency failures with destroying and secondary faults.

Repairs and maintenances are made in appropriately scheduled time with spare parts that are ordered exactly for that situation. That process would seem perfect in theory if it were real in practice. Things could be as it was mentioned only if all machines and all their parameters and emitting signals were monitored, and of course under the condition that all damages follow shapes able to warn us some weeks or months before the machines fail.

It is true that is difficult and very expensive to monitor the steps made by all the machines with sensors. In addition, we should take into account that sometimes the machines do not give us the appropriate type and sizes of signals as we wish.

The art of forecasting is recommended to monitor a machine, with the suitable sensors and organs, quite often, so that we detect how the fault develops, in function with operating time and prevent the unpredicted damage and its consequences. Obviously, the issue is also financial, because we should balance the financial benefit with the chance (risk) of failure.

It is advisable to ask about what we want to do to be sure when a machine will fail. We examine each machine and decide which will be out of monitoring and which will be monitored with sensors and organs suitable for each one and then select the frequency of measurements.

So, some machines will follow the philosophy of repair maintenance i.e., they will operate until they break down because they are not financial beneficial to suffer Condition Monitoring. With some other machines the minimal possible monitoring will be done with the criterion of taking into account the risk of failure and the cost. Finally, in some very important machines we may put a permanent monitoring system, designed for a 24-hour measurements and signals taking.

There is a serious inability on predictive-forecasting maintenance. If we just monitor the machines and maintain and repair them when required, we have essentially done nothing to improve the machine and increase its lifespan for the next stage. We just take the warning signs of the oncoming fault, and we fatally wait, by predicting approximately when the fault will happen and manage the maintenance before we enter adventures.

If, according to our notices, we could change the way of ordering and buying a new machine and the way of operating and maintenance in order to become more reliable and with a longer lifespan, then actually the maintenance cost would be much slighter. But this philosophy does not belong to predictive maintenance but to a more developed kind of maintenance as the next one.

Advantages of predictive maintenance

- Unscheduled faults are few
- Spare parts are ordered only when necessary (STOCK reduction)
- Maintenance is made only when necessary

Disadvantages of predictive maintenance

- High expenses for organs, systems, long hours and specified staff are required.
- The increase of the machines' lifespan is not ensured.

Precision maintenance

This type of maintenance is called like that (it is shown also as Proactive or Precision or reliability-based maintenance or Design Maintenance) because it uses precision methods of the technique to give the best result. Only experienced technicians can apply this method, while its philosophy is "make it once and correctly."

If we use a predictive maintenance program, we may get a fault signal from a bearing and of course we are going to order a spare part by scheduling the repair. This is right and is done. But why has the bearing broken? Not which point of the bearing is broken and gives the sign, but what caused that? If we knew the answer and if we could remove the cause of the bearing's damage, then the machine with the new bearing would have a longer lifespan after maintenance.

By that method of maintenance, we do not expect the machine just to undergo the fault. We perform activities to reduce the possibility of failure. We search the root of the damage, which means the main reason of fault cause and correct the problem in its root. The definition of this reason of the problem is a difficult issue, whose solution requires study of the procedures involved from specialized techniques.

Generally, this study includes:

- Research of historical elements and monitoring achievements of the machine.
- Performing tuning tests.
- Elements for the purchase and installation of the machine.
- Research of the archive of purchase and use of lubricants.
- The book of machine maintenance.
- The book of faults, etc.

Precision maintenance is the best way to increase the lifespan of a machine. Of course, we use the monitoring of the operating conditions of a machine techniques as other methods do in order to be achieved. But, at the same time, we use technologies required (e.g., study of forces on cogwheels, strength test of the various components, etc.) to determine the main cause (the weak working position) of each problem, in order to actually increase the reliability of the machines. What can a non-specialist do here?

Although there are many advantages of precision maintenance, at the same time there are also lots of obstacles to its adoption. The most important obstacle is that the staff must be trained and acquire a real experience on the research of the main cause of the problems based on the Mechanics, tribology, the precision measurements with the use of organs and devices, etc.

Advantages of precision maintenance

- The machines' lifespan increases
- The machines' reliability is greater
- Fewer failures and smaller secondary damages take place
- Very few unpredicted shuts
- Reduced maintenance expenses

Disadvantages of precision maintenance

- Increased cost of organs, systems, services and specialized staff
- Staff with increased qualification is required
- Enough time study of the machines is required
- A change on management mentalities is required (from TOP-DOWN)

Monitoring and control of the machines' operating conditions

Here we will refer to the reasons why we monitor the operating conditions of the machines, and we will see briefly the technologies used. The monitoring of operating conditions is a process in which we learn the machine's condition and check if its health and efficiency are constant or getting worse. It is clear that the monitoring of the operating conditions, which should not be confused with precision maintenance, is directly related to its health and efficiency, which are very important elements. Now, how this information for the machine is used as a part of the production and maintenance process is a completely different story.

Rotating machines emit lots of signals as their condition and efficiency gets worse. We should know how to receive and decrypt these signals. Although there are also cases where a machine is very quickly destroyed without giving an important warning, in most of the cases we receive enough signals for the oncoming fault and usually a long time before.

Rotating machines are like the human body (it subdues its complaints) and most of the maintenance departments treat the machines as many people treat their health.

Some people lead hard lives with activities (such as smoking, drinking, etc.) which are bad for their health. They ignore the signals their body emits related to the danger to their life. Thus, they usually suffer a lot of damage and if someone is lucky and they catch them, maybe the damage will get fixed. This reminds us of repair maintenance.

Still others, who lead a hard life, take vitamins and have substandard summer vacations thinking that this will eliminate their mistakes. They sometimes try to do the right thing usually after some symptoms and try again after other symptoms of damage to their health. This is similar to preventive maintenance.

Some people who are not fanatical followers of proper nutrition and healthy living, however, often go to the doctor who measures their blood pressure, performs blood and heart tests, and gives them medication if they diagnose any damage. Here we chased the symptoms instead of the reasons that caused them, and this case is similar to predictive maintenance.

Finally, we have people who play sports, live right, and know what is right and what is wrong through the information they have either on their own or from experts. For them health is a way of life and not a struggle. In addition, they visit the doctor and do examinations in order to stay healthy. This is like precision maintenance.

Monitoring the operating conditions of a machine is what the definition states. In the case of rotating machines we can control the shock pulses and vibrations of the machine in different positions, the motor current, the flow and the pressures of the lubrication network, to measure the temperatures of the bearing etc.

The first task of maintenance technicians is to take the measurements that represent the operating conditions and then to correlate the results. It is obvious that each measurement can reveal only part of the truth about some damage, while other measurements give no information about specific problems, e.g., the presence of metal particles in the lubricant give no evidence of the problem of poor balance. Combined monitoring is the complete monitoring of operating conditions.

It is necessary to pay attention to the conditions under which some problems arise, e.g., the tuning of the machine is not a problem because all machines are tuned, but if we have tuning in the operating speed then great vibrations will develop, which can lead to damage to the bearings and that is a problem.

An important element is the fact that the best results, for a machine's fate, are obtained when we compare the new measurements with the previous ones. It would be good to have permissible limits for each type of measurement. We must remember that simply monitoring the operating conditions of a machine does not increase its reliability, because simply knowing the operating conditions cannot change the reliability of the machine if we do not perform precision maintenance.

We must first understand well the different types of faults and how each one is identified with the measurements through the technologies used to measure operating conditions. These technologies include:

- The measurement of shock pulses.
- The measurement of vibration intensity.
- The method EVAM for the analysis and evaluation of the signals.
- The analysis of the lubricant's properties.
- The analysis of the metal wear particles in the lubricant.
- The analysis of the temperature field of the machine, etc.

Here we are going to deal with the first three. We will refer to the signal analysis, which can be used at predictive maintenance as well as at precision maintenance, as we have the opportunity to learn if a machine fails and determine the cause of the fault.

What is expected for the future?

More use of the Internet and tighter cooperation among Technology and the Condition monitoring, process monitoring, automated diagnostic systems and on-line systems is required in the future. Emphasis on information against data. We have noticed that we cannot have a complete image of the fault situation only with one technology. Analyzers and technicians need a knowledge of many technologies (multi skilled).

It is known that some years ago some big companies admitted the importance of improved maintenance and fault diagnosis and its effects financially.

So, in the future, the separating line between condition monitoring and process monitoring will not be clear. Today, the on-line monitoring systems are becoming more and more famous as they increase their abilities, while the diagnostic systems acquire greater importance.

On-line monitoring systems can be used on remote working positions. Not rarely, they are used in positions that send messages to the maintenance manager or the machine manufacturer as a part of the warranty process (e.g., wind generators).

It is certain that on-line systems have become more chic and with more possibilities, but also cheaper. They have become smarter, too. Instead of transmitting data megabytes from the measurement position at the central system, there will be a process to the point, and the diagnosis results will be transmitted to the central system.

The purpose is to reduce the operating cost (or increase the profit) through the maintenance cost reduction and through the minimization of expensive production stops (which means not to have production loss). This is achieved with the increase of the machine reliability and by controlling the maintenance schedule. This is achieved by making the necessary changes to secure that the facilities will be more reliable.

We should learn the reason about the machines' failure (high vibrations that create wears due to unbalance, poor alignment, tunes, and lubricants with polluting particles, inappropriate lubrication, bad quality repairs, and spare parts of low quality, designing and constructing problems etc.).

We should also check the machine to define problems in their genesis, in a way that we can program and carry out the maintenance in the most appropriate and most effective way in terms of cost time.

These goals can be achieved everywhere. They are already done by some industries around the world. Resoluteness is required from the management and suitable training of the technician staff.

9.3 BASIC KNOWLEDGE ABOUT VIBRATIONS

We begin with the basic knowledge: about the way that we receive the measurements, what we measure, the shape of the signals, how we will read the diagrams from the final results. Basic knowledge about vibrations includes: time waveform and vibration spectrum.

When we measure the vibration-shock on a machine's bearing, we essentially measure the answer of the hull of the bearing to the forces that are developed inside the machine. We should not forget the shock that is transferred to a machine from its environment. These forces are connected with all the rotating parts such as: the shaft, the rolling bodies of the bearings, the flaps of the fan. The shock measurements will show: if there is unbalance to the fan, poor shaft alignment, poor fastening of the machine to its shoes. Therefore, it makes sense that an experienced technician who will take the measurements, can conclude if something is wrong with the machine. How a technician concludes about these measurements? There are two basic rules. Rule number 1: he is able to recognize immediately diagrams, standards or ways of shock changing that correspond to specific faults. Rule number 2: as a fault is getting worse, the machine breaks down and the type of the tense of the shock changes (the tense increases, the type can change in many ways).

Everything started when the technician listened the noises of the machine through a screwdriver that he placed with the wood on the bearing's hull. Then vibration meter came, which measures an overall indication of the shock level. It is more sensitive than an ear, but it cannot distinguish what damage the shock comes from. New vibration meters appeared later, which focus their measurements on specific frequencies (usually high frequencies) to detect faults of the bearings. Such devices are still used: Shock pulse, Spike Energy, HDF etc.

In order to understand well the faults of the machines we must examine the shocks with every detail.

Let's consider an engine. If a load didn't exist (vacuum operation), if the bearings had zero friction, if there was a perfect alignment and if gravity didn't exist, then shocks-vibrations wouldn't exist. Yet, real conditions make a rotating shaft create shocks. A machine creates shocks because it has inner forces, which are due to: unbalance, poor alignment, bent shaft, etc. the shock measured is mainly due to these forces but also to the moving masses and the stiffness of the machine.

In order to understand the shocks, we should see the things from the side of the machine, with the help of an example. Let's consider of an axis and a new fan with eight blades that rotates with the axis of its engine. If we touch the engine's hull we may not feel anything, because we have supposed that the system is well. Now, let's assume that we place a small mass on one of the blades. At low speed we may feel nothing, when we touch the engine, as this mass is too small related to the mass of the fan, because, at low speed the small eccentric mass creates a slow centripetal force. Let's suppose that the eccentric mass is quite big. Even at low speed we feel the difference at the messages sent on the hull of the engine. Obviously, we feel a temporary annoyance with the same frequency of motion of the eccentric mass which is the same with the fan. (E.g., if the fan rotates with a turn per sec, then we will feel the annoyance every sec.)

In addition, the position of annoyance to our hand is identified with the position of the eccentric mass, which means if our hand is on the upper edge of the engine then we will be annoyed when the blade with the eccentric mass is at the upper position. This motion of the mass in function with time is simply harmonic. Motion is a sine wave.

The maximum rate of the wave takes place when the eccentric mass is up, while the minimum rate happens when the mass is down. The sine wave introduce the way of a vibration's change with time. This waveform is one of the most important chapters to the study of shocks.

The period of a waveform can be measured as the time between two maximum rates. Frequency waveform is the reverse of the period. If at the previous example the fan rotates about a circle per sec then the period is per sec, while the frequency is a circle per sec, which means 1 Hz. If we geminate the fan's speed and make a graph of the motion with the new sine waveform, then we have 2 turns per sec, which essentially is the frequency of the waveform, while the period of motion is the reverse, 0.5 sec. Waves are closer to each other. If we refer to time of 1 min, then the speed of 2 turns per sec is 120 turns per min. CPM = Cycles per Minute.

The rotation frequency is usually mentioned in Hertz (Hz), 1 Hz = 1 cycle per sec = circle/sec.

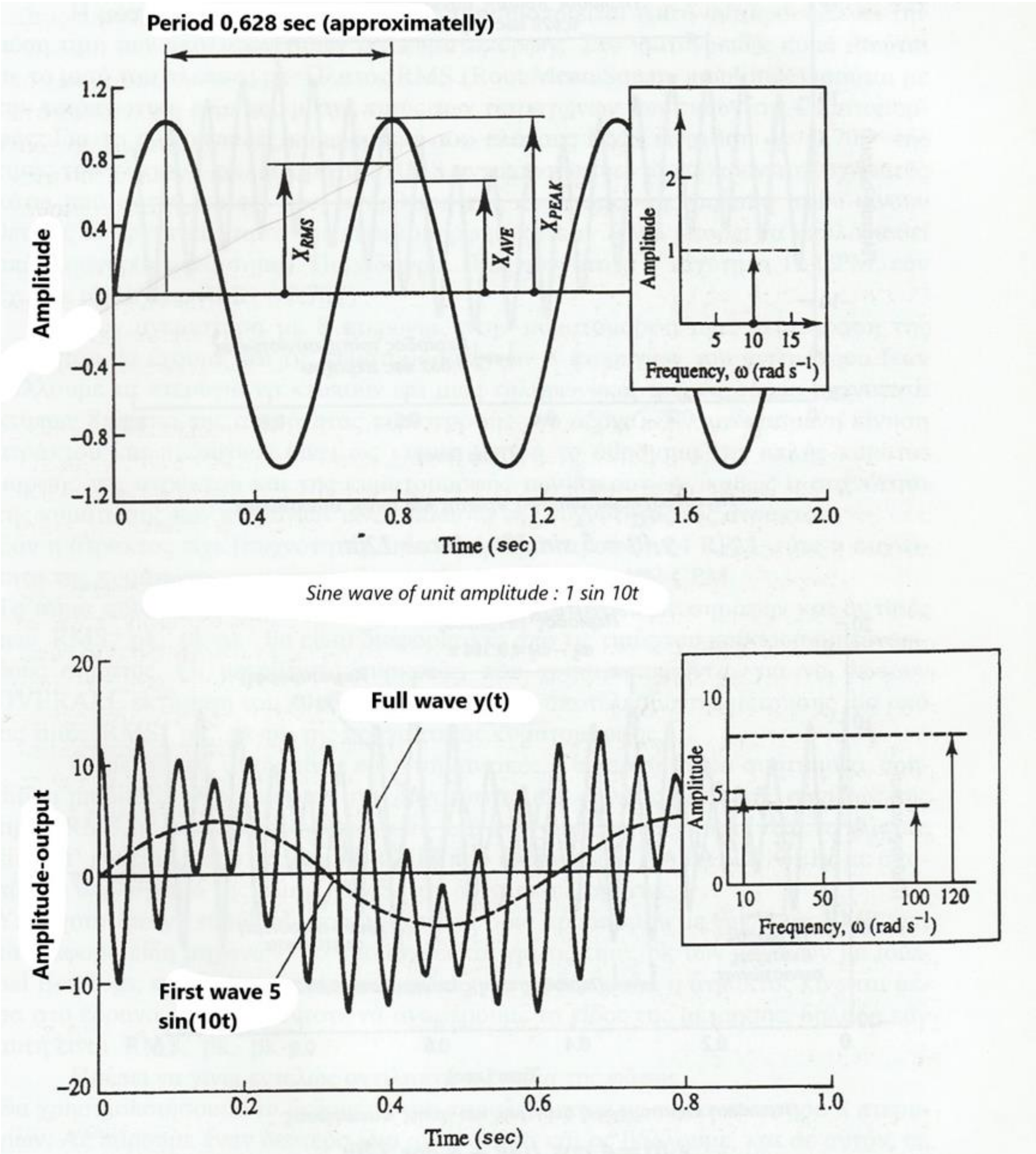
At the last example frequency is 2 Hz

The expression RPM = Revolutions per Minute is used for the study of rotating machines.

Notice: frequency units Hz and CPM are generally used for any frequency such as frequency of a sound, of a machine's noise, etc.

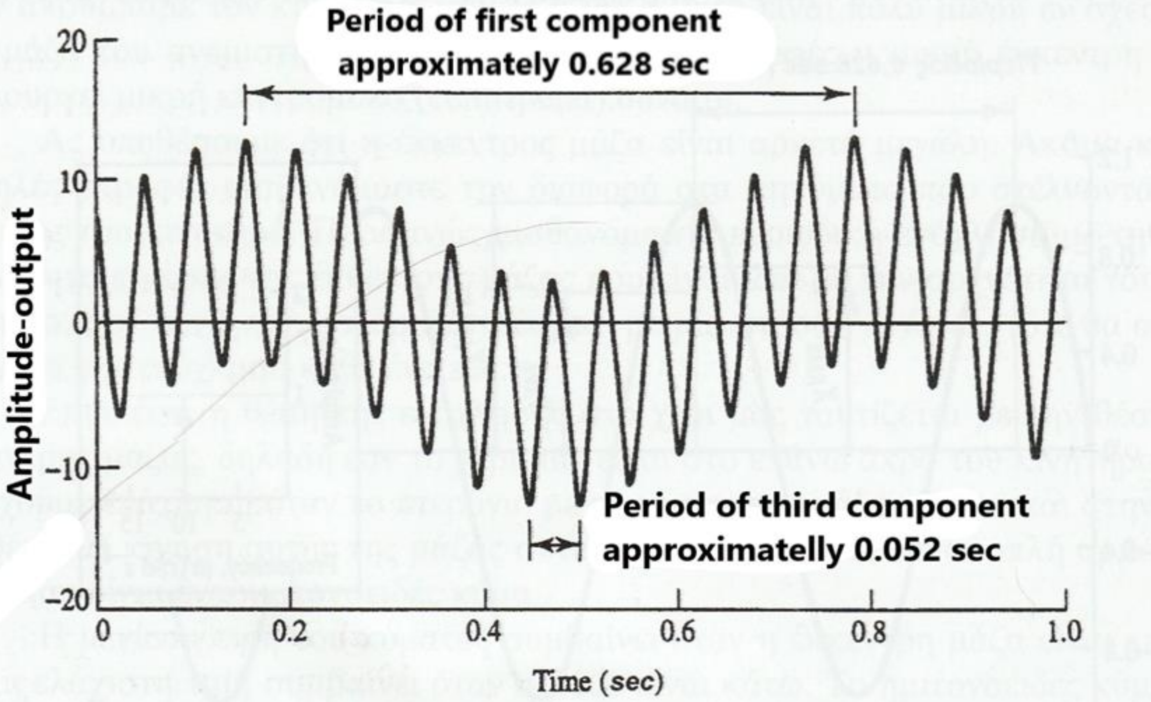
9.4 WAVEFORM

We will consider the effect of velocity on the waveform. As the velocity increases, the frequency increases, and the period decreases. Consequently, the distance of the ripples decreases (ripples = the circles of the waveform).



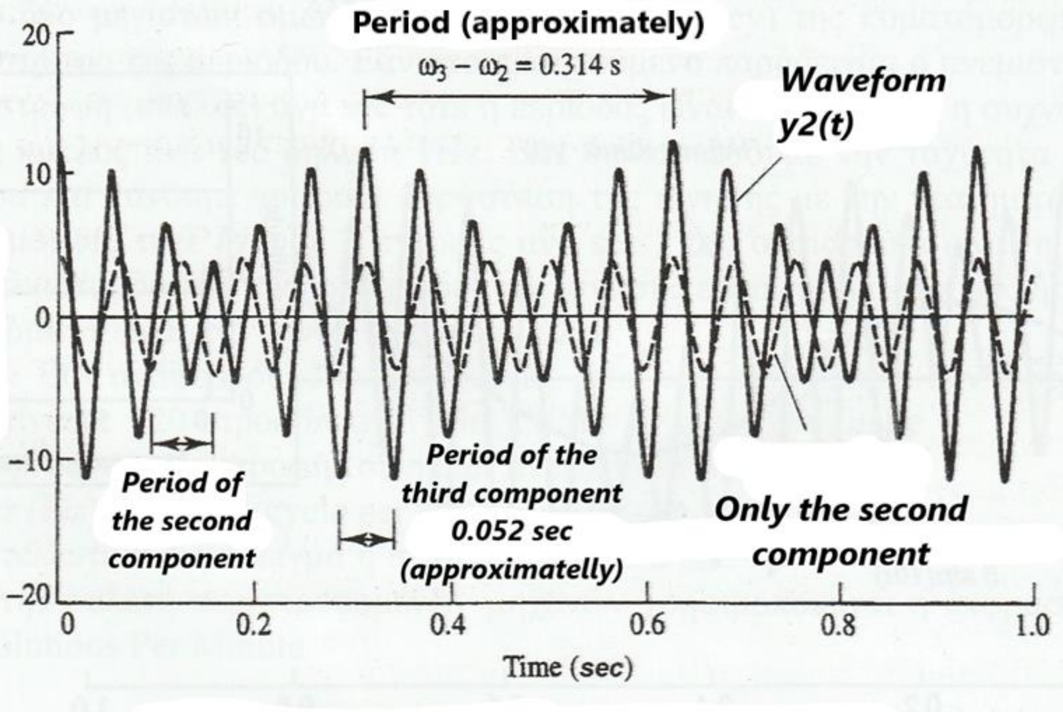
Periodic waveform: y (t) = 5 sin 10t + 4 cos 100t + 8 cos 120t

The distance of the hilltop from the depth of the valley represents the intense of the vibration and determines the amplitude of the oscillation. The amplitude of the oscillation from a hill to a valley is called Peak to peak (pk-pk). The amplitude of the oscillation (peak = pk) is the maximum distance from the initial position



Periodic waveform of first and third component

$$y_1(t) = 5 \sin 10t + 8 \cos 120t$$



Periodical waveform of the second and third components:

$$y_2(t) = 4 \cos 100t + 8 \cos 120t$$

to the one (to the hill) or to the other (to the valley) side, which of the two is greater as exactly happens to the real vibration signals. At sine wave we have pk-pk = double of the pk while in a real vibrating signal the amplitude of the hills may be larger than the amplitude of the valleys.

Average is rarely used. This represents the average of the absolute rates of the waveform. A sine wave is equal to the half of the amplitude pk. RMS (Root Mean Square amplitude) amplitude is equal to the square root of the average of the squares of the waveform rates. For the sine signal the amplitude rate is equal to 0.707 of the amplitude rate pk. RMS rate represents the area of the form under it (after the waveform rectification, which means after the negative rates of the waveform become positive, then the RMS can be calculated from the rectified system). Example: for velocity or frequency 12 CPM, if pk = 1 then RMS = 0.707.

In the fan with 8 blades, in the waveform of a rotation of the shaft we will also have the waveforms of the 8 fan blades (if we put the blades to knock on a phone card, we will see eight times the frequency of the axis rotation). The combined shaft and blades motion gives as a waveform the sum of the simple waveform of the shaft and the waveform of the blades, as the frequency of the blades ripple is eight times the shaft's frequency. If the shaft had (frequency) that is number of speeds 24 RPM than the frequency of the flutter ripple would be 8×24 = 192 CPM. This signal is the sum (superimposition) of two sine signals and the rates of RMS, pk, pk-pk will be different from the rates of the actual sine signal. The measuring devices that are used for an OVERALL estimation of the shock, give as a result of the measurement one of the rates RMS, pk, pk-pk of the real waveform.

These measurements are very important. Most of the condition monitoring systems offer the possibility of a direct measurement usually of rate RMS that is calculated either by the waveform, or the spectrum or directly through an additive CHIP that DATALOGGERS carry, with the purpose of calculating the dynamic signal's RMS rate.

There are international standards that define the shock limits with RMS rate for various types of machines. The same also applies for the pk rates of the machines with Journal Bearings, as pk rate is an identification of how well the shaft moves in the bearing. We must always mention the type of measurement that is if it is RMS, pk, pk-pk.

The concept of phase must be fully understood.

We will use the data of the example with the 8 blades fan. Let's get a second same fan and put a blade with the same mass on it and set the blades with the masses in the upper position. If we rotate the engines with the same speed, then both masses will pass by the upper position at the same time. If we touch with the one hand the one engine and with the other hand the other (same position), then we feel the annoyance of the masses at the same time, because the masses make pulse motions, have at the same time the maximum pk etc. Fans turn in phase. If the mass of one fan is up and the mass of the other is down, what is going to happen if we place our hands on the upper points of the engines? Every fan performs a pulse motion. The annoyance of each mass is maximum when it is in the upper point. Yet annoyances are not simultaneous.

Although each annoyance is made at every rotation, one is done after the other. At motion waveforms of masses and shafts, there a difference to the tune of the annoyances. Fans rotate out of phase. Here they have a phase difference of 180º. At the example we considered the phase only to one frequency, which is related to the rotation speed of the shaft. The bigger part of the shocks analysis essentially concerns the running speed and if anything happens in phase (at the same time) or with phase difference 180º (which means in opposite positions of the shaft).

9.5 DISPLACEMENT, VELOCITY, ACCELERATION

Until now we used our hands to “measure” the shocks. But actually, the measuring devices can measure three characteristic sizes: displacement, velocity, acceleration. A measuring sensor is placed at the position we placed our hand.

<u>Suppose the sensor measures displacement</u>

(The axis distance from the sensor)

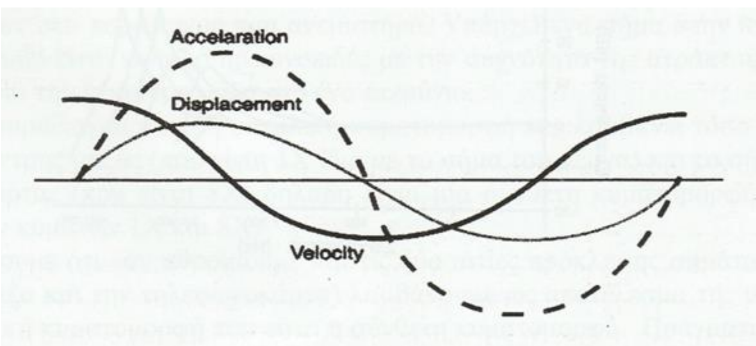
When the eccentric mass is in the lower point the waveform will be to the maximum (will show maximum removing) because the axis will be far away from the sensor due to the eccentric mass. This waveform represents the shaft’s removal from the vertical plane as it is measured from the displacement sensor. The displacement measurement is very important because it is proportional to the tendency developing in the bearing, to the retaining bolts, etc. The measurement of displacement on journal bearings shows if the axis is properly placed in the bearing or if there is contact between the axis and the bearing.

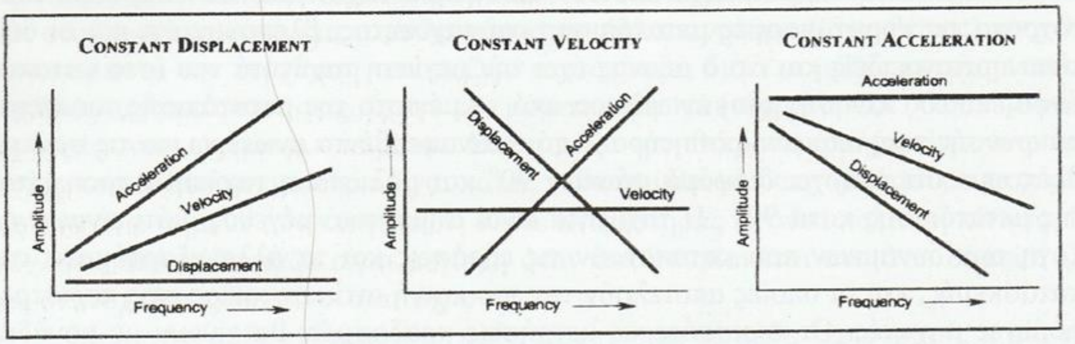
<u>Suppose the sensor measures velocity</u>

(The approach or removal velocity of the axis to the sensor). While the shaft moves on the vertical plane, its eccentric mass (maximum removal of the axis), which is at the lower point, begins to move to the sensor and when it reaches the upper point it starts to remove. The waveform now represents the motion velocity of the shaft on the vertical plane as it is measured from the velocity sensor. If we take the displacement and velocity waveforms at the same time, we notice that both of them are sine, and the axis has its maximum velocity (at its vertical plane of motion) earlier than the maximum of its displacement (removal) from the sensor. From what is mentioned above about the phases, we notice that there is a phase difference of 90º and in fact velocity precedes displacement by 90º. Velocity is an important size, due to the forces that strain the bearings and the other components of the construction, and which constitute the most common cause of failure in rotating machines. Most of the shock measurements will be done with velocity units.

Suppose the sensor measures acceleration

Beyond displacement and velocity there is also acceleration. If we observe the waveform of the velocity, we notice that the axis accelerates from the change of the lower position direction where it calms up to the middle position, where it has maximum velocity. There it begins to decelerate until it reaches the upper point, where it stops and changes motion direction downwards. The shaft during its downward motion follows the same acceleration process up to the middle position and the deceleration process up to the lower position. The greatest acceleration force is exerted when the eccentric mass of the shaft reaches the upper dead point and respectively the lower dead point. The acceleration waveform is sine but with a phase difference of 180º from the displacement waveform. By placing the three waveforms together we can see the phase differences. They are all important during the study of measurements and the signals. Acceleration is an important size because it is proportional to the internal forces of the bearings. As we will see to the measurements, acceleration G acquires great importance to high-speed engines, up to 120 000 CPM where displacement may be small, the velocity medium but acceleration can take great values. On the other hand, displacement (microns) is the best way of measuring in low-speed engines, under 100 RPM, where displacement may be high, while acceleration will be low. In middle speeds we get measurements of the velocity (mm/sec).




CONSTANT DISPLACEMENT
Acceleration
Velocity
Displacement
Amplitude
Frequency
CONSTANT VELOCITY
Displacement
Acceleration
Velocity
Amplitude
Frequency
CONSTANT ACCELERATION
Acceleration
Velocity
Displacement
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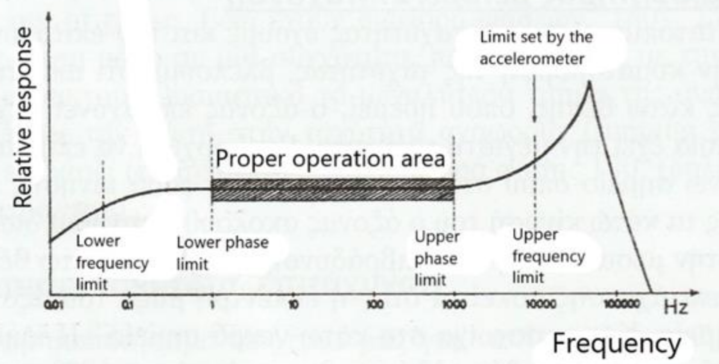
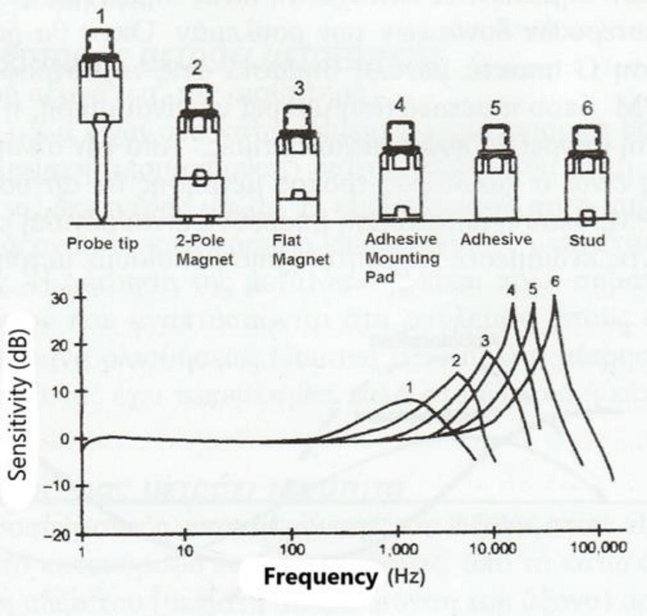
Frequency


Relative response
Limit set by the accelerometer
Proper operation area
Lower frequency limit
Lower phase limit
Upper phase limit
Upper frequency limit
0.01
0.1
1
10
100
1000
10000
100000
Hz
Frequency


1
2
3
4
5
6
Probe tip
2-Pole Magnet
Flat Magnet
Adhesive Mounting Pad
Adhesive
Stud
Sensitivity (dB)
30
20
10
0
−10
−20
1
10
100
1,000
10,000
100,000
Frequency (Hz)

9.6 THE FREQUENCY SPECTRUM

We know the waveform of the axis motion with the blades (if 1X is the frequency of the axis then for 8 blades the frequency is 8X). The final waveform consists of frequency 1X superposition of the axis and of frequency 8X of the blades. There are many shock sources in a machine, as the bearings, the cooling fan blades, the rods on the engine, coordination positions. These sources create different frequencies. The analysis of the vibrations is the art of noticing the changes on the waveforms and the investigation and connection of these changes with the changes in the engine.

In the example with the fan with the eccentric mass, consider of note down every year a waveform with exactly the same operating conditions, but with a greater wear and therefore with greater vibration, which means with greater oscillation width. As the frequency remains constant, the oscillation width increases, as someone has increased the amount of the eccentric mass of the previous example.

If we note down the total waveform (axis and fan) and if we assume that the waveform of the axis has no changes from measurement to measurement, then the changes are exclusively due to the fan. These are like they come from the greatest resistance of the phone card on the blades of the fan.

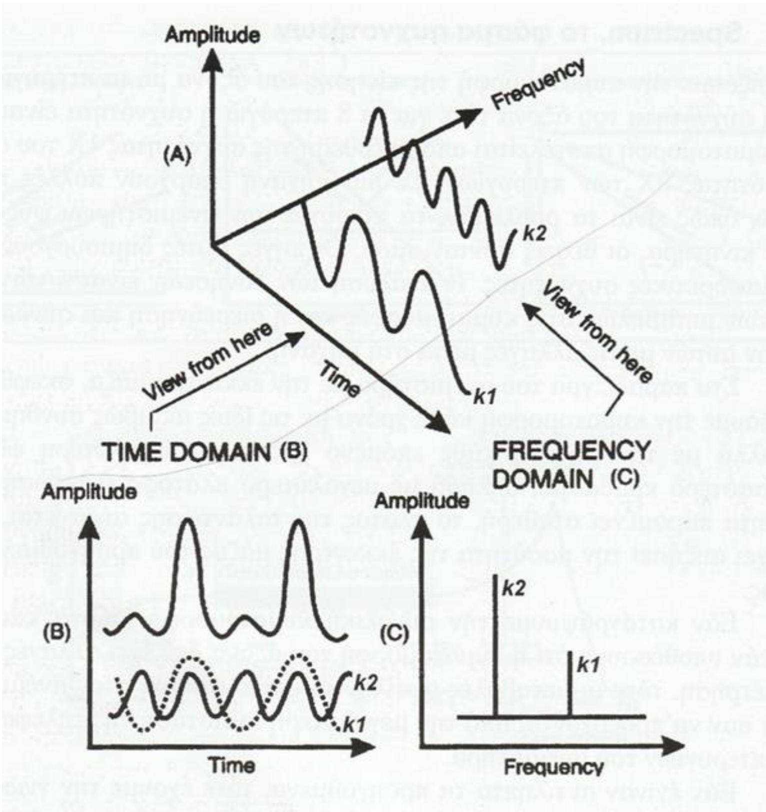
If the previous were perceived, then we could have the knowledge of the content of the vibration analysis, i.e., we can recognize the changes to the intense and the shape of the vibrations and diagnose the faults of the fan.

The waveform is very useful to the data analysis. It allows us to precisely see the way of vibration change from one moment to the next. If an impact load occurs (e.g., bearing rolling body knocks in a puddle in the internal ring of the bearing) we expect to see a spike in the waveform and increase of the impact pulse at the same time.

For the meaning of spectrum let us remember the waveform signals that exist in the example with the fan. There is a signal in the waveform, which is clear sinus with frequency of the shaft, as it resulted from the eccentric mass on a blade.

At this example, the total waveform includes the signal of the eccentric mass (that is the same 1X with the axis signal) and the phone card signal (that is 8X) that is a composite waveform as a sum of waves 1X and 8X.

We notice that if we sum both of the causes of signal induction (the eccentric mass and the phone card) we receive as a result of the superposition the final waveform which is composite waveform. Actually,



we can sum the two signals and the result of that superposition depends on the frequency, the amplitude, and the phase of the two signals. Here each signal has its own frequency and amplitude.

Waveforms are very useful. However, as many sources contribute to vibration, the direct fault diagnosis (from the waveform) in a machine is difficult. In the example above things are clear. If something changes, we can immediately find what is wrong.

The vibration study is also made in another way, through the spectrum. Spectrum results from the waveform through the procedure of transformation FFT = Fast Fourier Transform.

In a graticule system the horizontal axis includes the signal amplitudes. We must remember that a shaft has a great frequency when it has high speed.

Let us consider on the frequency axis 0 as a start (zero vibration) and end in 10 Hz (which means 10 circles per sec or equally 600 circles per min = 600 CPM =600 RPM).

Axis Y is shown in vibration amplitude and even we 0 as a start and 6 as an end (we will consider the units later).

At the example's fan we have added the eccentric mass and even that the frequency is 1 Hz = 60 CPM. Let us assume that amplitude is 3. We keep the same frequency 1Hz in the fan, but we place a bigger eccentric mass and even that we measure amplitude 6.

The axis of the fan (with the 8 blades) rotates with 1 Hz frequency. We will use the phone card, but we will remove the eccentric mass. We can hear the sounds of the blades as they hit the phone card with a frequency 8 times the fan axis' frequency, but the wave's amplitude is small even 1.

Without changing the rest of the characteristics, if we press the phone card deeper in the blades then the amplitude of the wave will increase, it may even become 2. What happens is obvious. While changing the vibration sources the waveform changes proportionally. Either the amplitude changes (the height of the line in axis Y) or the frequency changes (the horizontal position of the signal) or both of them. That is spectrum i.e., the frequency spectrum of a waveform. The previous example shows the procedure of spectrum creation, during which the total waveform is analyzed to its components, the frequency and amplitude is defined for each one and the spectrum is created afterwards.

The plane of time-amplitude represents a waveform, in the tree dimensions time-amplitude-frequency, for a given time, as for given time the plane of frequency-amplitude represents the spectrum. We have mentioned the frequency of the blades with the expression 8 times the frequency of the axis (which is relative frequency) and not with an absolute number in Hz or CPM. If X is the frequency of the axis, then the frequency of the axis is mentioned as 1X peak, as the frequency of the blades is 8X peak.

The absolute value of the frequency X can take any rate (1 Hz or 0.5 Hz or 20 Hz). We will use the absolute rates of the frequencies when we need them for their calculation.

However, we will better use in spectrum the relative frequencies that resulted from dimensioning (when the absolute frequencies have been divided to the frequency of the axis). In real machines with many components (so many vibration sources) the total waveform and the spectrum are much more complicated.

Let's repeat some significant points.

Every rotating component, is a source of one or more signals in a given time, in given frequency, amplitude and phase.

The combination of all signals (that we can take with sensors in a specific point of a machine) creates the total waveform. This may include the noise, vibrations from other machines and signals that result from other signals. Each waveform is analyzed at the clear signals that make it up, each of which has its own characteristics (frequency, amplitude, phase). We can in this way create the spectrum.

In spectrum each line by the axis Y represents a signal. Horizontal position represents the frequency of the signal, while the height of the line represents the amplitude of the signal.

Frequency can be in absolute units of rates such as Hz or CPM. Frequencies can be divided to the basic frequency of the machine and the relative frequencies to be written on the frequencies' axis. Therefore, the basic frequency, running speed, will be in line 1 and we call it of order 1. This procedure is called order normalizing.

When we analyze a machine and calculate the speeds of each component, these speeds are the expected frequency of each peak.

We repeat that the fan with the eccentric mass and the phone card, as in the example, has two frequencies in which peaks are expected in the spectrum. These are in positions 1X and 8X.

9.7 FAULT FREQUENCY PENDING

Waveforms and spectrums are based on the rotating components of the machines. We find the positions in the spectrum where picks can happen. It is not sure they will happen, but if a fault exists in a spectrum's position where the peak will appear, it is predefined. We should learn to study the machine and how to calculate these frequencies where the positions of the peaks are in the spectrum. These frequencies are called forcing frequencies = fault frequencies.

We will follow an example.

Let us assume that a shaft rotates with 1000 RPM. The frequency of the shaft is 1000 CPM and the expected peak position in the spectrum with absolute frequencies is 1000 CPM.

In the spectrum with the relative frequencies the peak position 1X is in 1. Let us now assume the cooling fan with the 8 blades. The frequency of the flutter will be 8X = 8000 CPM. If a problem with the flow or the wear on a blade exists, the peak position 8X in the spectrum will be 8.

We can now add a centrifugal compressor (impeller) to the previous system. We will calculate the frequencies in peak positions (forcing frequencies). The vanes of the compressor cooperate as well as the blades of the fan.

Given that the compressor has v = 12 vanes. Because the shaft of the engine is directly (without decrease or increase of speed with a gear or belt system) connected with the shaft of the compressor, they have the same speed 1000 RPM. Therefore, the relative frequency of the compressor's shaft is 1X and of the compressor's blades 12X.

We have three interesting frequencies:

1X for the shaft with 1000 RPM or CPM.

8X for the blades (b = 8 blades) of the fan.

12X for the blades (v = 12 vanes) of the compressor.

Any fault on the blades of the compressor will develop a peak in frequency 12X. If an alignment problem exists to the fan and the compressor a peak will develop to the basic frequency 1X. If we have a problem (flow or wear) to the blades of the fan a peak will appear to frequency 8X.

The calculation of forcing frequencies is made to help the fault diagnosis of the rotating machines. We detect the spectrum in the positions where the peaks will appear (that is the frequencies that create vibration problems).

9.8 FREQUENCY PENDING FOR GEAR FAULTS

We will analyze the use of cogwheels to the step-down and step-up speed transformers (gearbox = cogwheels box). The existence of a cogwheel box to the transmitting system enters additive frequency peaks (due to the cooperating gears). Yet the most essential effect of the box is that the output shaft has different speed than the input shaft. For a gearbox with two wheels (one level gearbox)

1 = the first wheel = to the shaft input

2 = the second wheel = to the shaft output

Number of teeth on wheel 1 = Z_1

Number of teeth on wheel 2 = Z_2

Speeds at the input of the shaft = n_1

Speeds at the output of the shaft = n_2

The relation n_2 = $n_1 Z_1 / Z_2$ is valid.

<u>Even the example</u>

If we remember the example of the engine that rotates with X = 1000 RPM, which has a fan with 8 blades and turns a compressor with number of vanes = 12. Let us interpose the previous one level gearbox between the engine and the compressor.

If $Z_1 = 12$ and $Z_2 = 24$ for $n_1 = 1000$ RPM input speed, the peak frequency at the output will be $n_2 = 1000 \cdot 12/24 = 500$ RPM

We will have the shaft of the compressor rotate with 500 RPM. The shaft of input to the gearbox rotates with 1X.

The output shaft from the gear rotates with 0.5X.

The peak frequency of the fan has not changed because this frequency is 8 times the frequency of its shaft, so it remains 8X.

The frequency of the flutter of the compressor with 12 vanes will be 12 times its shaft, but the shaft has a frequency 0.5X so the flutter of the compressor will have a peak frequency of 12·0.5X = 6X.

Conclusively, the frequencies are:

1X = to the shaft of the engine

0.5X = to the shaft of the compressor

8X = due to the flutter of the fan with the 8 blades

6X = due to the flutter of the compressor with the 12 vanes.

Another way of notation is to name the peak frequencies.

1X = 1XM = to the shaft of the motor.

0.5X = 0.5XC = to the shaft of the compressor

8X = 8XM = 8XB = due to the 8 blades of the fan

6X = 6XC = 6XV = due to the 12 vanes of the motor.

Many times, on cogwheel boxes the relation of the input speed I to the output speed O is set by the shape of I: O. In that case the relation 1:4 means that we have a speed amplifier, and the output speed is four times the input speed.

(Nothing from the known of the transmitting relation has changed, just $i_{12} = 0.25 = 1:4 = 1/4 < 1$ that obviously refers to a speed amplifier. It is an issue of defining the speed transmitting relation).

The frequency of the gear process = Gear mesh is calculated by the fundamental relation of the gears $n_1 Z_1 = n_2 Z_2$

If, for example we had a gearbox with $Z_1 = 12$ and $Z_2 = 24$ the gear frequency is 12 times the frequency of the input shaft or 24 times the frequency of the output shaft.

<u>Let us gather all the elements of the example</u>

$Z_1 = 12$,$Z_2 = 24$, $n_1 = 1000$ RPM,

Peak frequency in the output is

$n_2 = 1000 \cdot 12/24 = 500$ RPM

Consequently, the gear frequency is

$n_1 Z_1 = 12 \cdot 1000 = 12000$ RPM

$n_2 Z_2 = 500 \cdot 24 = 12000$ CPM

The same in every way it is calculated.

In multilevel gearboxes the procedure of calculation needs attention, especially in the intermediate shafts. Many times a gear frequency is obvious as a peak to the frequency spectrum.

Frequently, due to the lack of elements of the gearbox, this peak helps as to calculate the transmitting relation in an intermediate step.

<u>Example of a two-level box</u>

In a two-level gearbox, the transmitting relations are usually written as: input: intermediate: output e.g., notation 1:8:17 means that from the input with 1000 CPM we go to the intermediate shaft with 8 times of the input frequency, which means

Input shaft speed = 1000 RPM

Intermediate shaft speed = 8·1000 RPM = 8000 RPM

Output shaft speed = 17·1000 RPM = 17000 RPM

Even that in a two-level box we have:

$Z_1 = 35$, $Z_2 = 13$, $n_1 = 1000\ RPM = X$

$Z_3 = 27$, $Z_4 = 15$

It is about a speed booster because the drive wheels are larger than their cooperating ones. The peak frequency in the output of the first step is $n_2 = 1000 \cdot 35/13 = 2692.3$ CPM = 2.6923X that is the frequency of the intermediate shaft which is the input on the second step.

$n_3 = n_2 = 2692.3$ CPM

The gear frequency of the first step is

$n_1 Z_1 = 35 \cdot 1000 = 35000$ CPM = 35X

The output speed of the second step can be easily calculated by the intermediate shaft that has speed

$n_3 = n_2 = 2692.3$ CPM = 2.6923X

$n_4 = n_3 Z_3/Z_4 = 2692.3 \cdot 27/15 = 4846.14$ CPM = 4.84614 X

The peak frequency on the second step gear, i.e. the wheels 3 and 4 will be 27 times the frequency of the intermediate shaft,

$n_3 Z_3 = 27 \cdot 2692.3 = 72692$ CPM = 72.692 X or

15 times the frequency of the output shaft,

$n_4 Z_4 = 15 \cdot 4846.14 = 72692$ CPM = 72.692 X

9.9 FREQUENCY PENDING FOR BELT MOVEMENT FAULTS

Belt movements are very common in industry.

Yet the gearboxes are also used for speed change during the motion transmitting and power transfer. Mechanical engineer that applies the precision maintenance knows the precise forms of calculation. At first glance, we can use the forms of the single level gear, with the difference that instead of the numbers of the teeth we should put the nominal diameter (pitch) of the pulleys.

Example with belt moving with two pulleys.

1 = the first pulley = in the input shaft

2 = the second pulley = in the output shaft

(Pitch) nominal diameter of pulley 1 = d_1

(Pitch) nominal diameter of pulley 2 = d_2

Speed in the input shaft = n_1

Speed in the output shaft = n_2

At first glance we can write $n_2 = n_1 d_1 / d_2$

Remember that in a gearbox or a speed amplifier step (with cogwheels or with pulleys) the large diameter is always the slow one.

9.10 FREQUENCY PENDING FOR BEARING FAULTS

One of the most important components on rotating machines is the bearing. The frequencies in which a bearing shows fault are calculated as following:

$$FTF = \frac{1}{2}\left[f_{in}\left(1 - \frac{B}{P}cos\varphi\right) + f_{out}\left(1 + \frac{B}{P}cos\varphi\right)\right]$$

$$BPFO = \left|\frac{N}{2}(f_{in} - f_{out})\left(1 - \frac{B}{P}cos\varphi\right)\right|$$

$$BPFI = \left|\frac{N}{2}(f_{in} - f_{out})\left(1 + \frac{B}{P}cos\varphi\right)\right|$$

$$BSF = \left|\frac{P}{B}\left[(f_{in} - f_{out})\left(1 - \frac{B^2}{P^2}\ cos^2\varphi\right)\right]\right|$$

N= the number of bearing balls

B= the diameter of the rolling body

D= diameter of motion of the rolling bearings center = average diameter between the outer D_{in} of the inner ring and the inner diameter d_{out} of the outer ring, $P = (D_{in} + d_{out})/2$

f_{in} = The shaft frequency which is normally the same with the diameter of the inner ring

f_{out} = The frequency of the outer ring.

FTF = Fundamental train or Cage Frequency = cage rotation frequency

BPFO = Ball pass frequency outer race = rolling bodies pass frequency from a fault of the outer ring.

BPFI = Ball pass frequency inner race = rolling bodies pass frequency form a fault of the inner ring.

BSF = ball (roller) spin frequency = rolling body rotation frequency = two times the frequency with which the rolling body hits the inner or the outer ring, both hits together make the rolling bearing rotation frequency.

Φ = the contact angle of the rolling bodies = the angle that forms the line that joins the contact points of the rolling body with the inner and the outer ring with the frontal plane of the bearing. In uncharged bearings or in small axial loads the contact angle is zero.

<u>Case of immobile outer ring</u>

According to this $\boldsymbol{\omega_{out} = 0}$ so the frequencies are

$$FTF = \frac{1}{2}\, f_{in}\left(1 - \frac{B}{P} cos\varphi\right)$$

$$BPFO = \frac{N}{2} f_{in}\left(1 - \frac{B}{P} cos\varphi\right)$$

$$BPFI = \frac{N}{2} \omega_{in}\left(1 + \frac{B}{P} cos\varphi\right)$$

$$BSF = \frac{P}{B}\, \omega_{in}\left(1 - \frac{B^2}{P^2}\, cos^2\varphi\right)$$

<u>Case of immobile inner ring</u>

According to this $\boldsymbol{\omega_{in} = 0}$ so the frequencies are

$$FTF = \frac{1}{2}\, f_{out}\left(1 + \frac{B}{P} cos\varphi\right)$$

$$BPFO = \frac{N}{2} f_{out}\left(1 - \frac{B}{P} cos\varphi\right)$$

$$BPFI = \frac{N}{2} f_{out}\left(1 + \frac{B}{P} cos\varphi\right)$$

$$BSF = \frac{P}{B} f_{out}\, \left(1 - \frac{B^2}{P^2}\, cos^2\varphi\right)$$

9.11 PHASES OF MEASUREMENT ANALYSIS AND EVALUATION

The analysis and evaluation of the measurements for fault detection is done by the following phases:

Discover = Detection phase

Analyze = Analysis / diagnosis phase

Detect the cause = Root cause analysis phase

Verification = Verification phase

In most of the cases we will have gathered a great number of data and measurements, such as: measurements of total shocks intense, special measurements for bearings (HDF, spike energy, shock pulse, etc.), spectra, waveform, demodulated spectra, etc.

In an ideal condition we would have an Expert system or an artificial intelligence system that would be able to search the data and tell us which machines have problems, what kind of problems each of them has, what must be done for each machine, which components are needed and how the precise maintenance calculations will be made so that nothing from the above will happen again.

If this is what you think is happening, you have not understood it well. Although you will have enough assistance from devices and calculating systems, the rest expert system is each of you that will deal with precision maintenance.

Firstly: each person should be able to collect measurements and analyze. This requires knowledge of the theory, the measuring devices, and the machines.

Secondly: you must gain experience to make suggestions for machine maintenance. Most of the experience will be gained on your own in your job and do not forget the well-intentioned and professional cooperation with the other technicians.

Thirdly: you must have organized all of your activities really well to use your time effectively. Every time we measure with data collector we return with a huge number of data, from which we receive the waveforms, the spectrums and anything else that the device produces. We need lots of time to analyze each measurement, which does not exist.

On the other hand, the most important the machine is and the slower the measurements are done, the bigger the chance of losing the warning signs of a fault is.

So, what are we going to do?

First of all, we need a first control system that will process the new data and show us which machine may have a problem.

This system must not forget any machine that has a problem, while it must not show us as problematic a machine that is not (false alarm). That is the first phase, Detection phase, during which we discover what we need i.e., the list of the machines that probably have a fault.

With this list given, Analysis phase starts, which means the analysis procedure starts during which we determine the nature an intense of each machine's problem. This phase requires careful analysis of the data, i.e., special measurements (discovering of the phase difference, coordination, transient effects, lubricant analysis, thermography, etc.) that are not grind measurements.

When we finish and diagnose the fault a technical report follows in which we can for example say that the operation of a machine can be continued in another speed or correct some damage etc.

The procedure of the detection of the cause of a fault follows, because according to the rules of precision maintenance that a mechanical engineer follows, the first question is why the fault happened.

E.g., a fault on a bearing happened. The bearing alone (if it is properly calculated and assembled and if it is properly maintained) has no faults. The fault of a bearing is usually due to unbalance or poor alignment, or poor lubrication. The question remains: why was the machine unbalanced or poor aligned or bad lubricated? That is how Root cause analysis is done. This consists of the base of precision maintenance and aims the elimination of the cause of the problem. It requires excellent knowledge of the industrial components.

We check the maintenance booklets for various things such as: if balancing has ever been done to the machine's components, how often these faults are. Finally, perhaps the operation of the machine itself develops loads that cause the problem. By following the precision maintenance and executing its commands, we ensure the final correction of the problem, with the purpose of increasing the machine's lifespan and most important increasing its reliability. We must not forget that a machine's maintenance is followed by childish diseases (with increased possibility of fault), therefore we must be sure that the machine will return to production without alignment, lubrication, sealing problems etc.

The phase that follows is the verification through which we verify that:

a. The fault was corrected.
b. The machine operates well again.

It is found at the top of the pyramid and refers to a minimum number of machines. The goal is for someone to be found as soon as possible, far away from the pyramid base. So, free, one can deal with fault diagnosis and ways of avoiding faults in the future.

The difficult point is if the fault diagnosis and its correction is done but the symptoms are still present. E.g., the shocks were attributed to unbalance of the component and after balancing the shocks are still there. The only hope is the unbalancing diagnosis was correct and in the meantime another problem appeared.

Conclusion. We do not make recommendation only with one measurement. We completely investigate the machine and, in all ways, (as the patient gets in the hospital, first is generally checked and gets to the surgery if necessary).

<u>Exercise</u>

Let us consider that you just came back from the data collection with the device in your hand. You connect with the computer and the SOFTWARE and transfer the content to the hard disk. A way of studying is someone to start reading the measurements for each machine. Although this is tiring, it is the only way to notice the measurements without relying on automatic reading from SOFTWARE.

SOFTWARE companies have taken the Exception Report, according to which we make the scanning on the new measurements, we compare them with the ALARM limits, and we compare them we the previous measurements and a list with the new results comes out.

The list contains the machines that failed, due to a problem and gives an indication for the seriousness of the problem.

The list shows the new measurement and the level of exceeding the permission limit, as a % percent.

(100% means that the new measurement is two times the limit (ALARM) or with a total number.

It may also contain the comparison with the previous measurement.

How is the limit defined? What tense rate makes us start thinking of maintenance? Up to which limit is the correction of the fault recommended?

It is not easy to set a limit and say: from that limit and above machines are maintained, while below that limit the machines keep on operating. This limit has many parameters that are based on the machine’s elements: size, load, importance, history.

For example, a machine cannot have the same shocks as a grinder. We are more interested in the high vibrations of our important and expensive machines than the smaller and cheaper ones.

There are two ways of defining the operation limits of our machines.

First way. We use published ALARM LEVELS and regulate the machine limits to constant rates (FIXED ALARM LIMITS).

Second way. We begin with existing vibration measurements and calculate the ALARM LIMITS.

There are advantages and disadvantages.

Some boards are presented for these limits, as they resulted from various sources (industries, military, and researchers)

Limits are based on the type of the machine defined by the load, the velocity, or the type of operation it executes.

ISO Standard number 2372/10816 gives acceptable instructions for machines with velocity 10 up to 200 Hz (600 up to 12000 RPM).

The previous limits refer to total vibration rates, usually for the area of vibration frequencies of 10Hz- 100 Hz.

But what happens when we are interested in a specific frequency? Then the constructors of the data collection devices have suggestions for solutions.

9.12 OVERALL SHOCK MEASUREMENTS WITH A VIB DEVICE

Each part of a machine oscillates in its own coordination frequency (Eigen frequency), which means that the machine vibration essentially includes many different frequency components.

The various hand implements when they measure shocks, they usually measure vibration severity based on ISO 10816, where it is defined that vibration severity will be the RMS level of the vibration velocity, measured on the frequency field 10 to 1000 Hz.

This requires a different type of sensor and different way of signal processing. Instead of measuring the amplitude of the transient wave to a high frequency, in the case of shocks the reading of the vibration severity represents the average rate of all vibration components in a wide range of low frequencies. **Vibration severity** is proportional to the level of the vibration energy of the machine and, consequently, it is a good indication of the destroying shock forces exerted on the machine.

Causes of vibration

Every moving machine has oscillations. In order to define the normal oscillation and shock level of a machine we must take into account:

- The type of the machine's operation and the forces involved.
- The stability or stiffness of its mechanical construction.

A big DIESEL machine oscillates more than a small electric engine, due to the different forces that develop.

A machine based on a solid concrete foundation oscillates less than a same machine which is bolted into a flexible metal frame, because the whole construction is more stable and stiffer.

Excessive oscillation in a new machine is a sign of innate structural weakness of the same machine of bad characteristics of its coordination. The increase of oscillation and shocks level from the level of the proper condition is basically due to three causes:

- Something loosed
- Something lost its good alignment
- Something lost its good balancing

Ways of shock measuring

The number of oscillations per second is the oscillation frequency measured in Hz (Hertz = cycles per second), (1500 rpm = 25Hz). The cyclic motion can be measured as displacement, acceleration, or velocity. The range of displacement is the distance to which an object removes and is measured in mm. A body that moves front and back continuously accelerates and slows down. Acceleration is measured in mm/sec^2 or in g (1g = 9.81 m/sec^2). The oscillation velocity changes constantly. We can measure its maximum rate, but an average rate gives a better indication of the forces that slip into the motion. Every VIB device shows the intense of oscillation that is defined as a rate of oscillation velocity RMS (root mean square) and is measured in the frequency field of 10 Hz to 1000 Hz.

Shock measurements in three directions

The points of shock measurements should be near the hull of bearing. The measured shock at the point of taking the measurement must represent the overview of machine shocks, which means that we must not take measurements at the points that are less stable than the machine's frame (as the protective frame and the bumper). The more measurement points we have the easiest we detect the special mechanical problems (e.g., if the main problem results that is the unbalanced fan, then the measurement of shock near its position is enough to detect it).

The common practice is to measure three shocks on each bearing:

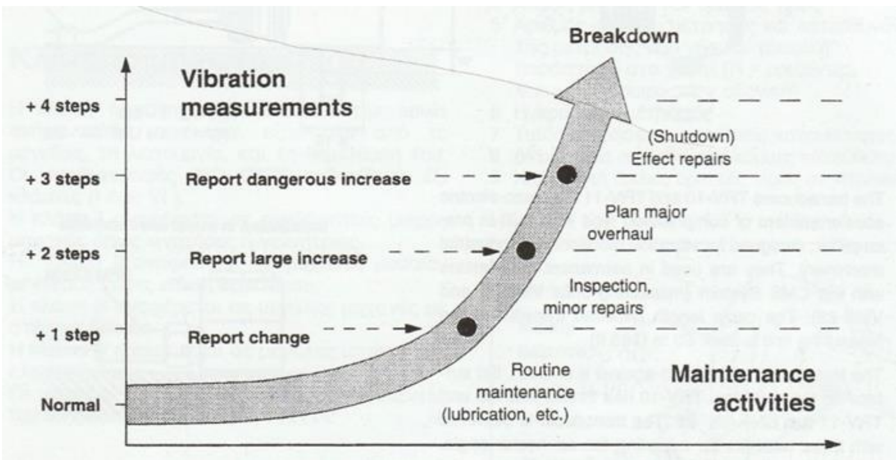
- Horizontal – H
- Vertical – V
- Axial – A

Axial shocks are usually produced by poor alignment of the shafts. Horizontal oscillation mainly represents the balancing problems. Vertical oscillation usually gives information about the mechanical weakness of the construction.

Pay Attention to vertical or other (not horizontal) shaft.

Diagram of shock measurements

The indication of real measurement of the shocks means minimal things for a non-insider.

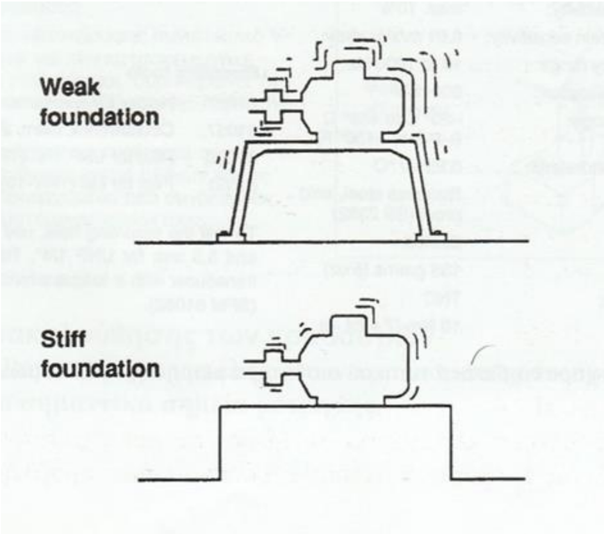


The mentioning of increasing the shocks per steps enables us to easier understand the necessity of preventive-predictive-precision maintenance.

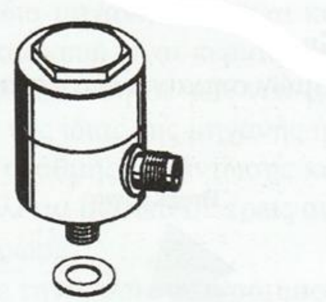
The increase of the shock intense per one step is considered to be important and must be mentioned.

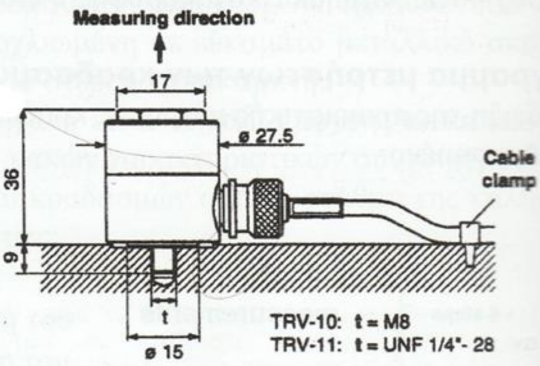
The increase of the shock intense per two steps must be investigated.

The increase of the shock intense per three steps comprises an important machine change and requires direct intervention.



Vibration Transducer TRV-10 / 11



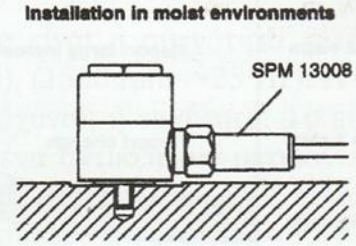


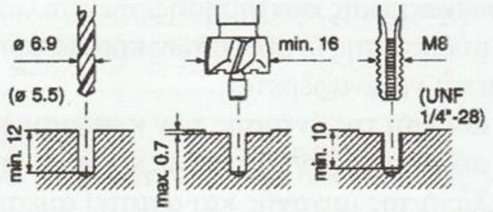
The transducers TRV-10 and TRV-11 are piezo-electric accelerometers of compression type with built-in pre-amplifier, designed for vibration monitoring of industrial machinery. They are used in permanent installations with the CMS System (measuring units VMS-22 and VMS-23). The cable length between transducer and measuring unit is max. 50 m (165 ft).

The transducer is mounted against a smooth, flat surface on the machine. TRV-10 has thread size M8 and TRV-11 has UNF 1/4"-28. The transducer is delivered with three washers for adjusting the connector angle. Each washer turns the transducer 90°. The coaxial cable (SPM 90005-L or 90267-L) with TNC connector must be secured with a clamp close to the transducer.

For installations in moist environments, it is necessary to use sealing TNC cable plugs SPM 13008 to prevent cable corrosion.

Installation in moist environments





Technical data

Measuring range:	0.2 to 100 mm/s 0.01 to 4.0 inch/s
Nominal sensitivity, main axis:	20 µA/mm/s
Transverse sensitivity:	max. 10%
Typical base strain sensitivity:	0.01 m/s²/µ strain
Linear frequency range:	10 to 1000 Hz
Max. peak acceleration:	600 m/s²
Temperature range:	–20° C to +55° C (–4° F to +130° F)
Typical temperature drift:	0.25% / °C
Casing:	Stainless steel, acid proof (SS 2382)
Design:	Sealed
Weight:	135 grams (5 oz)
Connector type:	TNC
Torque limit:	10 Nm (7.4 lbf · ft)

Mounting tools

81027	Holder for counterbore
81057	Counterbore, diam. 20 mm
81030	Pilot for UNF 1/4" (TRV-11)
81031	Pilot for M8 (TRV-10)

To drill the mounting hole, use drill bit 6.9 mm for M8 and 5.5 mm for UNF 1/4". Torque and unscrew the transducer with a torque wrench and a 17 mm socket (SPM 81092).

Sheet with the characteristics of a typical vibration sensor

<u>Evaluation of vibration level measurements</u>

VIB device is firstly used for the general evaluation of the machine's condition and is not dedicated for detailed fault analysis. Yet, by measuring various points and in different directions someone can have a proper indication of the fault type of a machine that causes the increased vibration level.

<u>Machine classes</u>

The class or category of vibration in which each machine belongs, depends on the size, the operation, and its foundation. ISO standards 2372 determine six classes (I to VI).

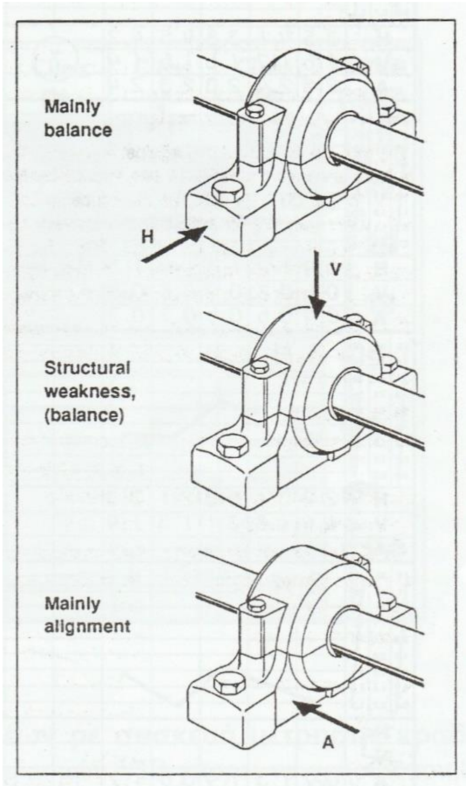
Class I refers to independent small machines such as engines or generators.

Class II refers to medium size machines without special foundation.

Class III refers to large machines with a stable base.

Class IV refers to large machines with elastic base.

Classes V and VI refer to heavy and reciprocating machines.



<u>Main fault analysis</u>

It is possible to make a simple fault analysis, by having various measurement points in a machine and by measuring to three directions. As a general rule, we have the following:

- The most representing measurement for charging measuring in a machine with alignment problem, is the measurement of horizontal component H.
- The measurement of vertical component V informs us about the constructor weakness of machine and completes measuring information about H for balancing.
- Axial measurement A refers to axis alignment problem and problems coming from joints and badly bending shafts.

<u>Clarifications</u>

(Of next page sketch)

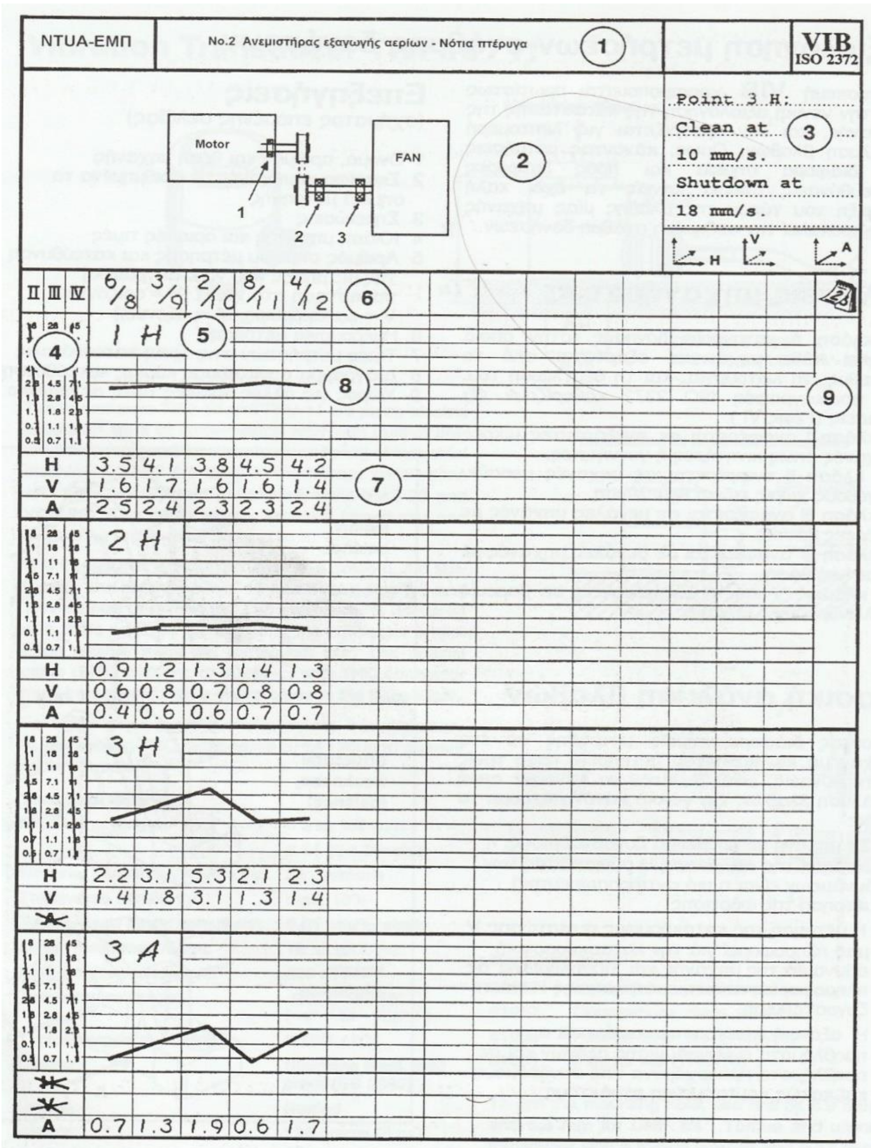
1. Name, number and position of the machine.

2. Machines draft with numbered measurement points.
3. Notes

1. Class of the machine and limit rates.
2. Number of measurement point and direction of the machine that becomes a chart on paper (H = horizontal, V = vertical, A = axial).
3. Date of measurement
4. Measurement rates on the three directions
5. Main direction component chart
6. Space for other limit rates in mm/sec.

<u>Shock monitoring card</u>

In follow-up form card we keep a measuring archive for shock monitoring **for the most important measurement points.**



Shock monitoring card

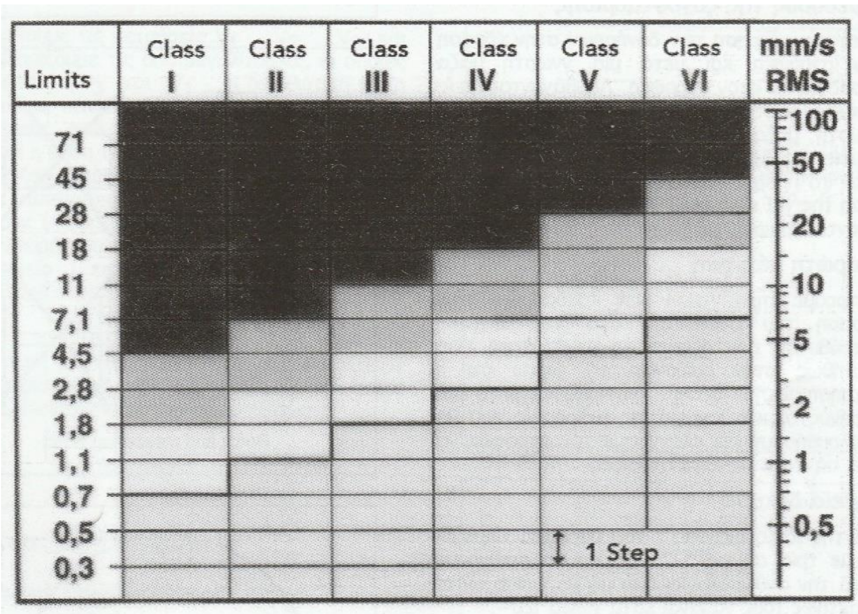
This card has note posts for tasks that must be done:

1. Number, name and position of the machine.
2. Chart of the machine with number at measurement points (1, 2, 3).
3. Notes
4. Class of the machine and shock limits for each one.
5. Measurement point and direction (H, V, A). Here 1H.
6. Date of measurement
7. Measurement rates in three directions (H, V, A).
8. Main measurement rates chart. Here 1H.
9. Space for other limit rates.

<u>Shock machine categories according to ISO 10816</u>

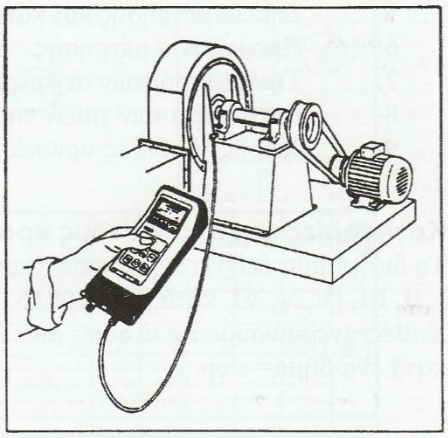
The chart shows the limit rates of shocks for the 6 machine categories I, II, III, IV, V, VI according to ISO 10816. Each step = approximately 1.6 times.

As we ascent classes I to VI the limit RMS ascents as well per one step = step.



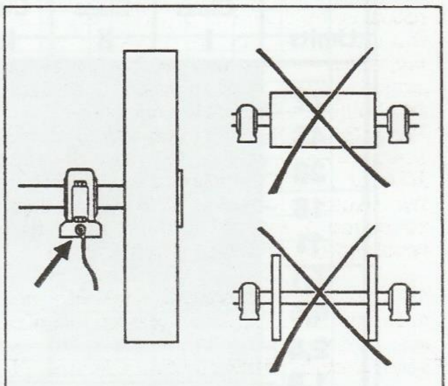
Balancing of one level carriers with shock measuring device VIB. A shock measuring device VIB can be used for balancing of one level carriers that have a significantly larger diameter than their width, e.g., as the most industrial fans are. The number of speed RPM must be greater than 600. The measuring point is the fan's bearing, and the measurement is made in the horizontal direction.

Balancing of one-level rotors with a VIB machine



VIB device can be used for a simple balancing with enough precision and reduce the shocks of the machines in acceptable levels.

This method does not require the rotor's dismantling and is suitable for machines with number of speed greater than 600 rpm and only for one-level rotors, e.g., rotors that have much bigger diameter than their width. Most industrial fans belong to this category.

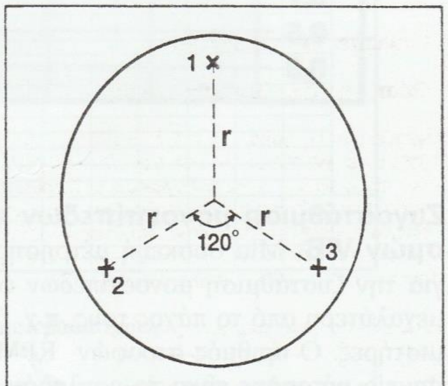


Rotor and measuring point

Before balancing is applied the rotor must be cleaned and its looseness or any broken or poor aligned components that create additive shocks must be fixed.

The balancing method

The vibration measurement is done on the bearing of the rotor and then a known mass is attached to the rotor. Three additional measurements are received, with the known mass placed in a point of the same radius each time, and in angle of 120º to the previous. The necessary mass and its position are graphically calculated for a successful balancing with a compass.



Trial points on the rotor

First measurement

We measure the shocks intense on the rotor bearing, to the radial direction that indicates the greater value (usually in horizontal H). Even that this measurement is the V_o. We use exactly the same point as well for the measurements that follow and check the rotor to always have the same speed.

Trial points

On the same radius r of the rotor, we note three points 1, 2, 3 that we number counterclockwise in such a way that their radius is to angle 120º.

Trial mass

Known mass M_t in grams is attached on one of the three points 1, 2, 3 each time and must be such so it can create enough bad balancing problem to the rotor.

Second to fourth measurement

At points 1, 2, 3 the corresponding shock measurements are V_1, V_2 V_3.

Calculation of corrective mass

Let's follow the example:

Trial mass $M_t = 12.3$ grams

Shocks without mass $V_o = 6.1$ mm/s

Measurement with mass at 1 $V_1 = 7.7$ mm/s

Measurement with mass at 2 $V_2 = 3.2$ mm/s

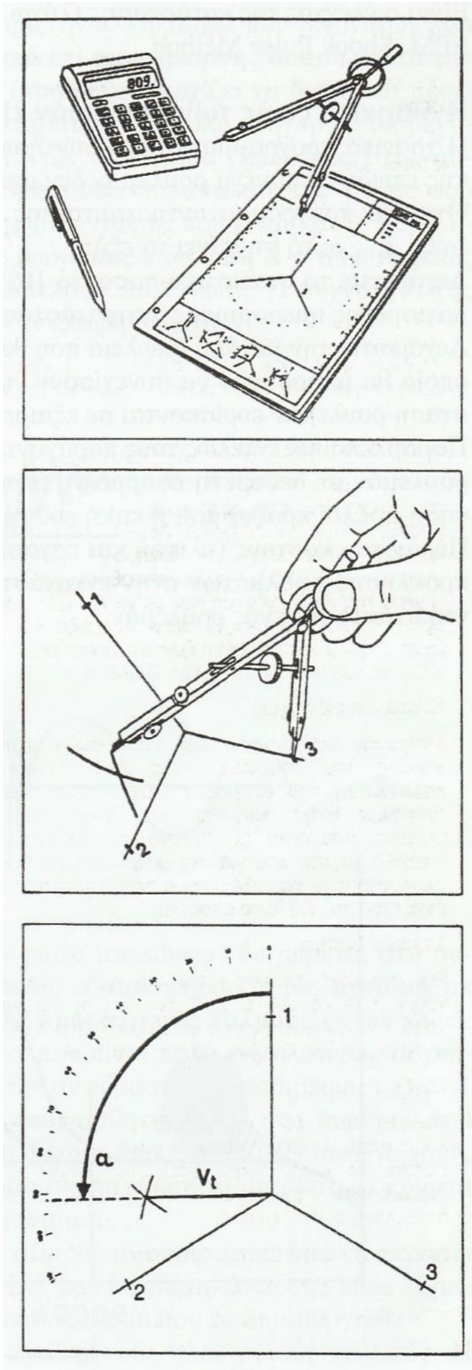
Measurement with mass at 3 $V_3 = 10.3$ mm/s

We search about the following rates:

V_t = Division factor

M_b = Corrective mass in grams

a = angular position of mass M_b



We choose the appropriate scale for the engraving on sketch sheet. We use a compass and take an opening equal to V_o (=6.1) and by using the right drawing scale we note the three points 1, 2, 3 on the drawing sheet counterclockwise.

We compare the measurements V_1, V_2 V_3 and define the two largest, which are V_3 and V_1 . The corrective mass always depends on the two larger measurements, while the third assist for the position of the corrective mass to be found.

We draw an arc of circle with center the point 3 and radius V_3 (as in the meanwhile we take into account the drawing scale) to the side of point 2. We do the same thing with point 1 and radius V_1 and so we define the section of two arcs.

By the drawing scale we calculate the distance of the section to the axis origin and find $V_t = 4.6$ which is used at the equation below:

$$\frac{M_t \times V_0}{V_t} = M_b$$

Results $M_b = 16.3$ grams.

Position of the corrective mass

We measure the angle between V_1 and V_t (always with these two and always from V_1 to V_t counterclockwise. The corrective mass of 16.3 grams must be set on radius r (as the radius that the trial mass was placed) and in angle a = 91^o from the radius of point 1 counterclockwise.

For precision insurance the distance between the section points and 2 must be V_2.

9.13 THE IMPULSE METHOD SPM

It is the control of the bearing condition (due to lubrication) with SPM (Shock Pules Method) method.

<u>The lifespan of a bearing is unpredictable</u>

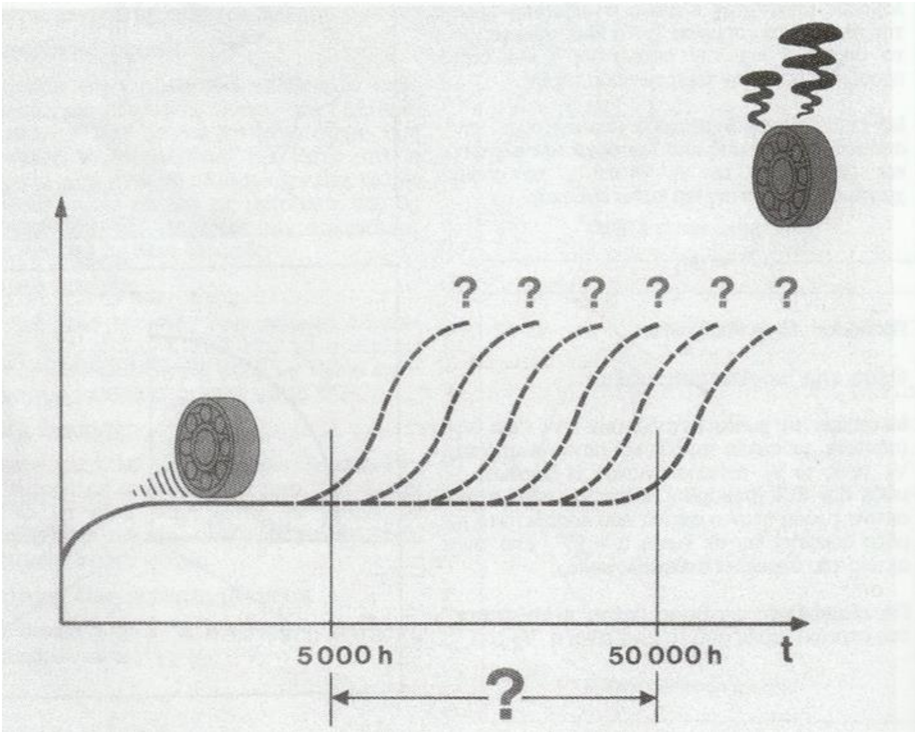
The timely scheduled maintenance (time-based maintenance) is deficient because no bearing can predict its lifespan. When the bearing replacement program has as a base lifespan L_{10} it means that:

We accept the statistical percentage of 10% failure of the bearings before the scheduled date of their replacement.

We accept the great loss in lifespan of the bearings which could continue operating. Many of the bearings to be replaced are in excellent condition.

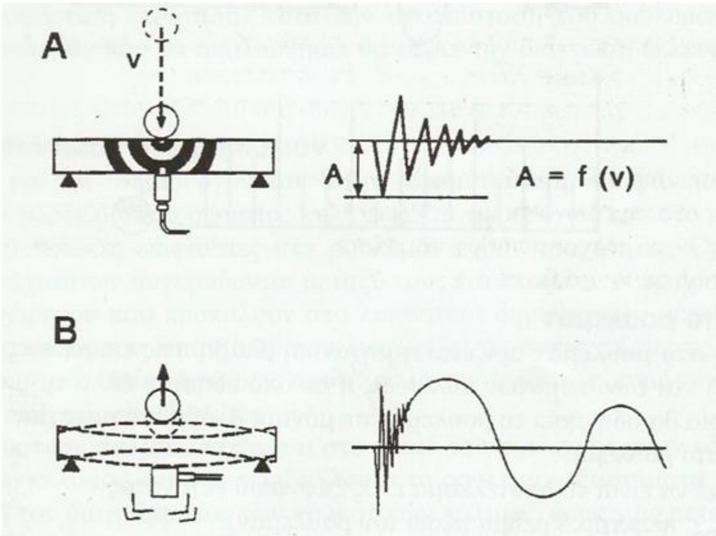
We completely overlook the factors that shorten the life of a bearing in a given application such as: poor lubrication, excessive vibration-many shocks, the axis misalignment etc.

Beyond the cost (material and work) there is also the chance of creating-causing faults to installation or in the machine during replacing a bearing.



Difference between impulse and shock

The measurement of Shock pulse from the measurement of vibration. Let us consider the sequence of events that take place when a metal sphere hits on a double beam. At the time of impact, a pressure (impact) wave starts to diffuse through the material of each body. The wave is transient, and it attenuates very quickly. Similarly, when an impact wave hits the transducer of the Shock pulse device will cause wane oscillation of the mass of the transducer. The first range or amplitude A of oscillation which is the larger is a function f of impact velocity v that is A = f (v). Soon after, both bodies begin to oscillate. The oscillation frequency is a function of the mass of in contact bodies.

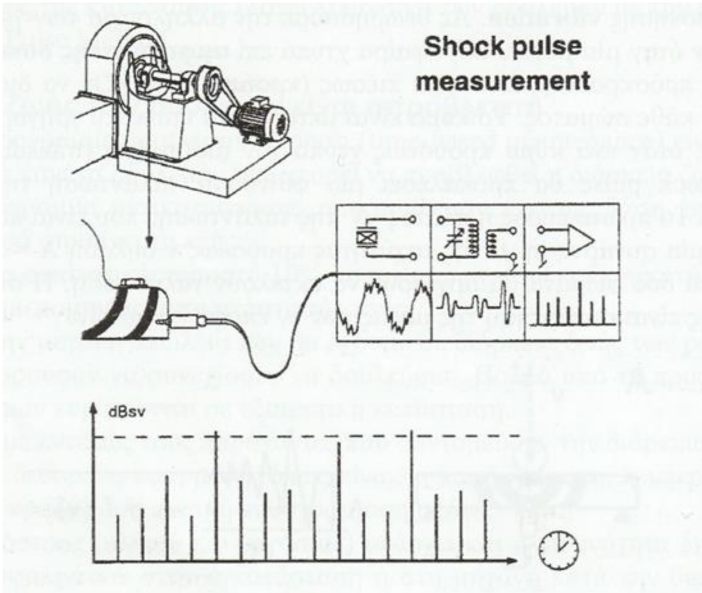


Impulse

A shock pulse transducer stimulates in coordination frequency of 32 kHz. Consequently, this transducer reacts with a large frequency amplitude to weak shock pulses. The rest machine vibrations that are of lower frequency, are filtered and are not part of the transducer reaction. The symbol of the transducer appears in the first box and in the box below this the vibration signal which comes out of the machine which is under examination, which contains the machine’s vibration and in superimposition the transient waves of the shock pulses at a coordination frequency.

The electric filter appears in the second box which allows a series of transient waves at 32 kHz to pass by. The oscillation amplitude of each wave depends on the energy of the impulse that created it.

The transient waves convert into analogue electric pulses. The last box contains the impulse of the bearing that was converted into a sequence of strong and weak electric pulses.



Damage to bearings

A **damage to bearings** is defined as the mechanical damage on the surfaces of the raceways of the rolling bodies or at any other part of the bearing, which will lead the bearing to a constant damage and failure.

The damage to the bearing:

- May be the result of a sudden event

(E.g., electric current through the bearing)

- It may occur when the bearing is on stand-by

(From vibrations or rust corrosion)

The surface disaster usually develops gradually after the passage of a great time span. Yet, this process can be boosted by various kinds of direct or indirect lubrication fault.

Operating conditions

The form of a bearing's shock pulse represents its operating condition, which is the sum of all the factors that affect its functionality during the machine operation. The measurement device does not recognize the causes on its own. The retrieval of the cause for a yellow or a red indication of the device requires the evaluation process of the measurement.

Bearing lubrication

Only a small percent of the bearings gets destroyed by **natural** fatigue of the steel. In most of the bearings the material fatigue starts very quickly because the rolling bodies and their raceways are not suitably separated with a protective layer of lubricant.

Direct causes to lubrication fault are:

The deficient or poor lubricant supply to the bearing.

The usage of wrong lubricant type.

The existence of contamination to the lubricant.

In addition, indirect faults, can be considered to be any effects that do not create a direct problem to the bearing surfaces but affect the lubricating layer, such as poor axis alignment and the increased preload (motion) during the installation of the bearing. Then the fault of the bearing that will follow will be diagnosed as lubrication deficiency.

Lubrication deficiency on bearings

Shock pulses are created to the rolling interface of the charging rolling body and its raceway on the bearing. These rolling interfaces on bearings have roughness. The tops of this roughness crash each other and therefore roughness creates (due to the inseparable ones that cause to the lubricant) changes of the pressure of the lubricant layer that separates the cooperating surfaces. Both of these causes have as a result the development of pressure-shock pulses waves.

These shock pulses are dispersed to the bearing material, to the bearing housing, to the hull and the surrounding machine components of the bearing. The form of the shock pulses chart, includes weak and strong pulses. The intense of a shock pulse is measured on sound scale of DECIBEL in **dB** units.

$\boldsymbol{dB_c}$ is the carpet value, that represents the measured number of the weaker pulses on the chart. This carpet value is the level of that pulse intense achieved from about 1000 pulses per second. The carpet value is found with the headphones when the rapid sequence of the sound pulses is converted to a continuous sound.

This corresponds to a somewhat lower frequency (200-300 pulses per second) but the difference to the value of shock pulse is insignificant.

Characteristic values for the lubricant layer

The low carpet value of DECIBEL is directly related to the width of the lubricant layer on the rolling surfaces of the bearing.

The carpet value is low when the rolling bodies and their raceways surfaces are completely, or almost, separated by the lubricant layer (i.e., the lubricant layer is constant and does not break to allow the metal-to-metal contact).

The carpet value increases when the width of the lubricant layer decreases and so the duration of metal to metal contact increases to the interfaces of the rolling bodies with their raceways to the bearing.

Factors that affect the width of the lubricant layer

In a given appliance of the bearings some of the factors that affect the lubricant layer are constant and cannot easily change for maintenance. These can be:

- Static and dynamic load.
- Geometry (carpet, hull, bearing, housing, shaft)
- Rolling speed
- Necessary preload

Some other factors that can be affected, by making possible the **lubrication** streamlining that is the lubricant layer, and consequently the increase of the bearing lifespan. These can be:

- Change of preload (especially when relies on a wrong placement)
- Axis alignment
- Total load (due to alignment and preload)
- Proper lubricant supply to the bearing
- Suitable type and quality of the lubricant

The bearing temperature depends on **constant** factors (velocity, environment, load, etc.) and the **lubricant**.

Maximum DECIBEL value

Maximum DECIBEL level represents the measured number of shock pulses. If the maximum value is high and at the same time the difference between high and low value is great, then this is caused by the faults that exist in the interfaces or from the existence of foreign particles among the rolling bodies and their raceways (i.e., foreign particles in the position of the lubricant or in the lubricant).

The growth of the fault

The changes on the operating conditions during the bearing's lifespan from its placement to its replacement are controlled with frequent measures of the shock pulses. The diagram shows the changes of **dB** for a bearing with regular increase of its lifespan (without placement faults and with satisfying lubrication). If these changes continue to exist on the shock pulse value, then these are associated to the faults of the bearing.

The scale is divided into three zones:

Red = poor operating conditions.

Yellow = defective operating conditions.

Green = proper operating conditions

An ordinary example of good operation comprises the form of the shock pulse that has

- Low carpet value $\boldsymbol{dB_c}$ (proper width of lubricant layer) and
- Maximum value $\boldsymbol{dB_M}$ inside the green zone (no faults on interfaces).

A remarkable increase of the development of the lifespan curve is a sign of a fault-damage beginning on the bearing. Maximum value $\boldsymbol{dB_M}$ enters the yellow zone and the difference between $\boldsymbol{dB_M}$ and $\boldsymbol{dB_c}$ becomes greater.

When the maximum value $\boldsymbol{dB_M}$ enters the red zone with a coincident increase of the difference between $\boldsymbol{dB_M}$ and $\boldsymbol{dB_c}$ and with the increase of the carpet value $\boldsymbol{dB_c}$ these show a visible fault on the bearing.

Curve of the bearing lifespan

Some changes to the indications are normal. They may be due to the difference of the temperature or the load changes or the difference of time from the re-lubrication or to other variables of the operating conditions of the machine or the bearing.

The abrupt arrows of new breaks cause high value indications which decrease when the fault is partly smoothed. Significant elements for the bearing replacement scheduling comprise the way of change and the percent of indication changes.

Bearings with indications on the red or the yellow zone must be checked more frequently than the green zone bearings. The plan below is recommended for the periods of measuring and check:

- Green zone = 1 to 3 months
- Yellow zone = some days to a week depending on the way that load conditions of the bearing etc. change.
- Red zone = frequent measurement, we should schedule the bearing's replacement. If the indications rise sharply, we must schedule the direct replacement of the bearing.

Systems of continuous monitoring

The systems of continuous monitoring for shocks and vibrations are used when:

- We expect a sharp change to the bearing or the machine condition.
- Sharply operating shuts that are expensive or dangerous.
- The manual bearing control is impossible due to the construction of the machine or due to the environment.

Adjusted shock pulse indications

The absolute shock pulse level in a bearing, measured in $\boldsymbol{dB_{sv}}$, depends on the rolling speed and the bearing condition. The device should be fed with two data, in order to neutralize the rolling speed effect on the measurement:

- Shaft’s diameter (mm or in).
- Number of speed (rpm).

Device T2000 calculates the initial value $\boldsymbol{dB_i}$, which is the origin point of the condition scale of the specific bearing. So, the condition scale is graded to regulated values of shock pulses $\boldsymbol{dB_N}$.

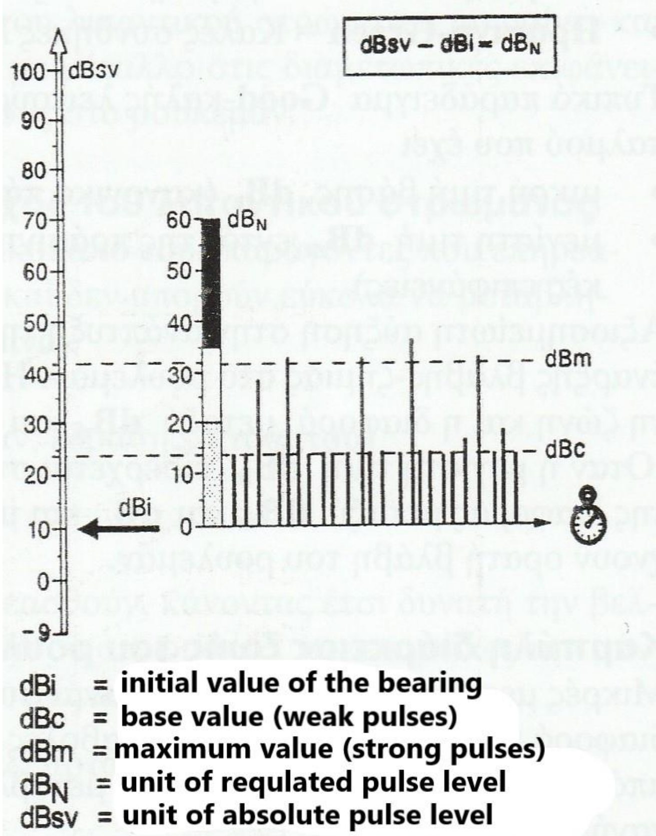
Device T2000 receives samples of shock pulse indications that occur during a time period and shows in the screen:

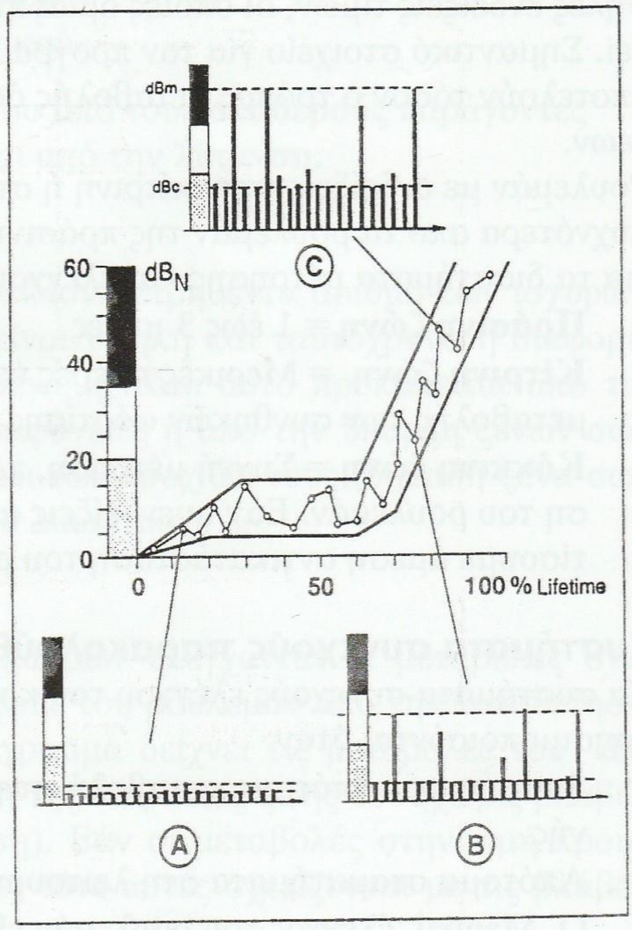
- The maximum value $\boldsymbol{dB_M}$ of strong shock pulses that are a few.
- The carpet value $\boldsymbol{dB_c}$ of the weaker shock pulses that are a lot.

The maximum value $\boldsymbol{dB_M}$ determines the bearing position in the condition scale. The difference between $\boldsymbol{dB_M}$ and $\boldsymbol{dB_c}$ is used for better analysis of the reasons that cause reduced or poor bearing condition.

Characteristic development

On a well condition bearing, the maximum values must be in the





green area and the difference between dB_M and dB_c must be small (case A).

By the passage of operating hours, the general shock level tends to increase slowly. The difference between dB_M and dB_c will increase with the existence of a small surface wear (case B).

Lack of lubricant will also set the bearing to the yellow area but in that case the difference between dB_M and dB_c will normally be very small.

A remarkable increase of dB_M shows great tendency to the material and wear start. Bearings with great wear have a high dB_M and a great difference between dB_M and dB_c (case C).

Shock pulse sensors

All three types of shock pulse sensors are connected to the same switch 9(SPM) of device T2000. The type of sensor that is going to be used depends on the kind of measurement point. SPM suggests the use of permanently installed pulsations in the position of the measurement point and the sensors with a quick link, for a systematic control of the shock pulses.

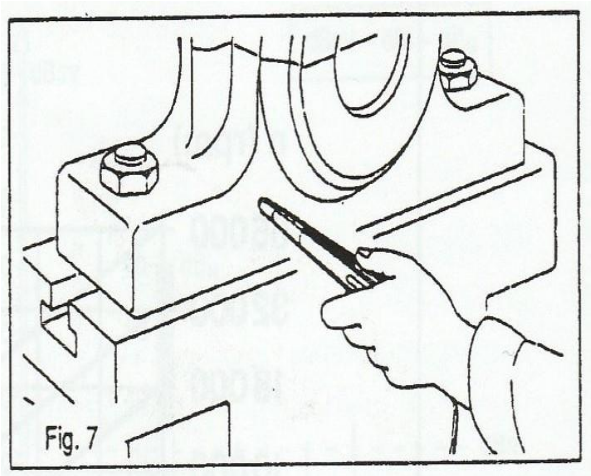


Fig. 7

Pulse tip sensor

This sensor is sensitive only to incentives in its axial direction. Its axis should point towards the bearing. The head of the sensor has a pretension with a spring and moves in a rubber housing. We press the sensor on the hull of the bearing, at the point of the measurement, until the rubber is tangent to the surface to achieve a constant pressure. We hold the sensor constant to avoid the sliding of the surfaces. The center of the sensor head should be tangent to the surface of the measurement point. We should avoid the use of

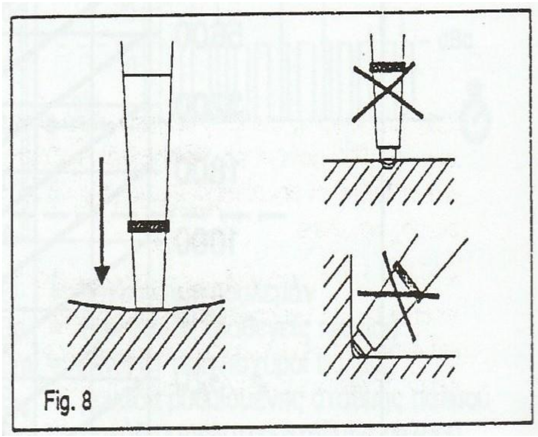


Fig. 8

sensor in cases of small pits and curves with radius of curvature smaller that the sensor's head (Fig. 8).

Sensor with a quick coupler

We should pay attention for the quick coupler socket head to be clean. We can use a few clean greases on that head to achieve better signal efficiency.

The quick coupler links the sensor's cable with the point of with low pressure and turn towards the right until it stops. We press (1) the quick coupler to the point of measurement (2) and we turn it toward the left, to remove it, Fig. 9. Sensor TRA-20 is used as in Fig. 9B.

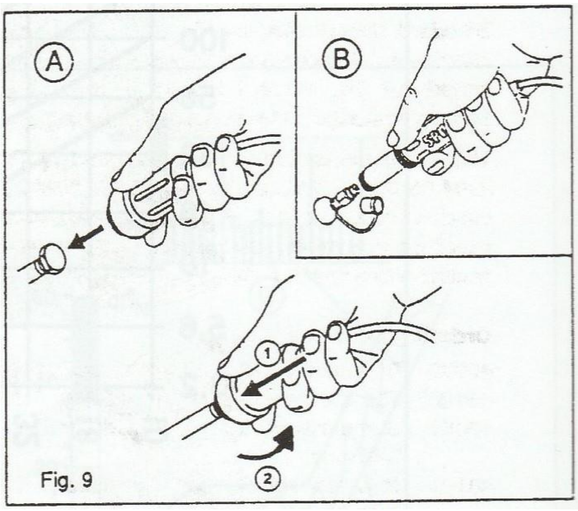


Fig. 9

Terminal measuring point

We connect the single-axis cable to the T2000 device and the terminal measuring point. We reposition the cover to the

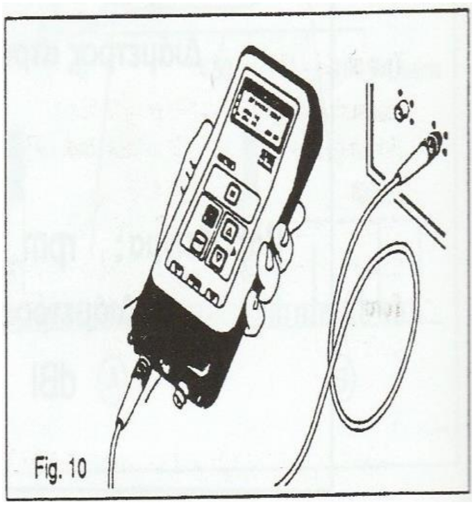
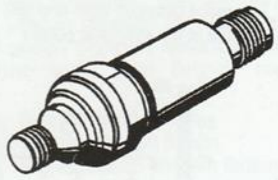
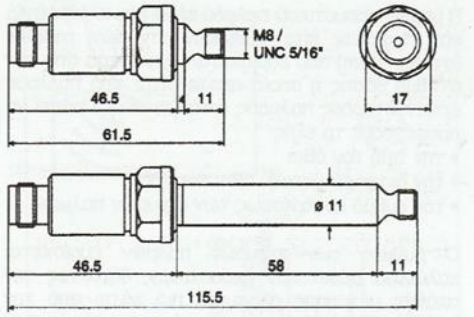


Fig. 10

terminal after the measurement for its dust protection (Fig.10).

Standard Shock Pulse Transducers



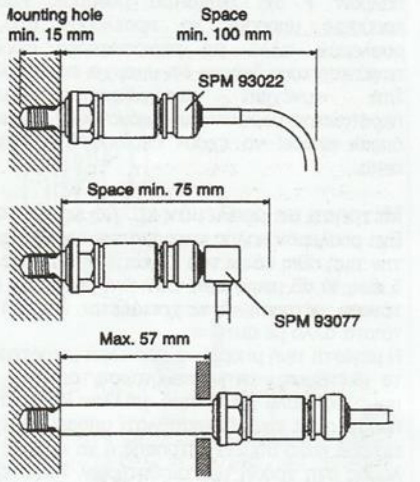


Standard shock pulse transducers are used in all permanent SPM installations for bearing monitoring. They are installed in countersunk mounting holes on the bearing housings.

A shock pulse transducer converts the shock pulses emitted by the bearing into electric signals. A coaxial cable connects the transducer with a measuring terminal or measuring device.

Transducer housing and base are made of stainless acid proof steel, suitable for aggressive environments. Standard thread size is M8, with UNC 5/16" as an alternative. Standard length (A) is 61.5 mm. A long transducer (B), length 115.5 mm, is used to reach bearing housings beneath protective covers.

The transducer is normally connected with a TNC plug, SPM 93022. A TNC angle plug, SPM 93077, can be used in narrow spaces. To prevent cable corrosion in moist environments, the coaxial cable must be connected with a sealing TNC plug, SPM 13008.



Ordering numbers

40000	Standard shock pulse transducer, M8
40100	Standard shock pulse transducer, UNC 5/16"
40001	Standard shock pulse transducer, M8, extended
40101	Standard shock pulse transducer, UNC 5/16", extended

Technical data

Measuring range	Max. 100 dBsv
Housing, base	Stainless steel, SS 2382
Design	Sealed
Temperature range	-30° C to +150° C
External overpressure	Max. 1 MPa (10 bar)
Torque	15 Nm, max. 20 Nm
Connector	TNC jack

Mounting tools

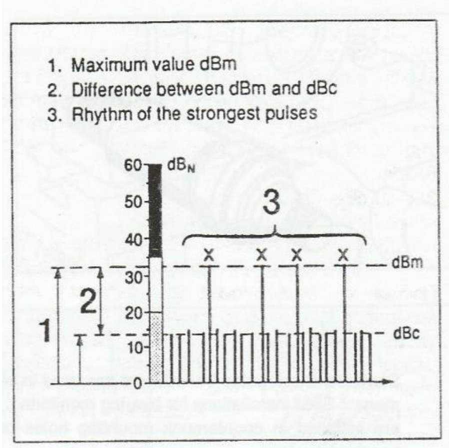
82053	Countersink with fixed pilot for M8
81027	Holder for countersink
81028	Countersink, angle 90°, 12 mm dia.
81031	Pilot for M8
81032	Pilot for UNC 5/16"

To drill the mounting hole, use drill bits 6.9 mm for M8, 6.6 mm for UNC5/16".

Torque and unscrew the transducer with a torque wrench and a long 17 mm socket (SPM 81092).

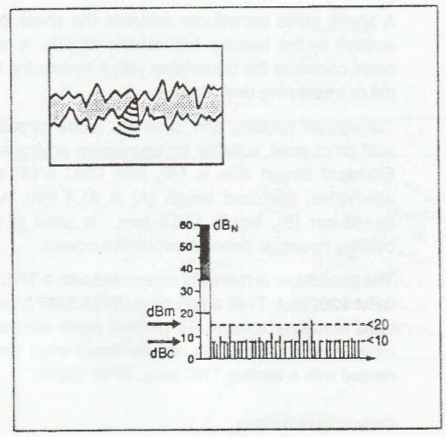
Standard forms of shock pulses to bearings

The form of shock pulses is a sequence of either random or rhythmic strong pulses ($\boldsymbol{dB_M}$ level) which is higher than the carpet level that consists of lots of wearer pulses ($\boldsymbol{dB_c}$ level). We have to watch out for the following:



- $\boldsymbol{dB_M}$ value
- The difference between $\boldsymbol{dB_M}$ and $\boldsymbol{dB_c}$
- The rhythm that strong pulses appear.

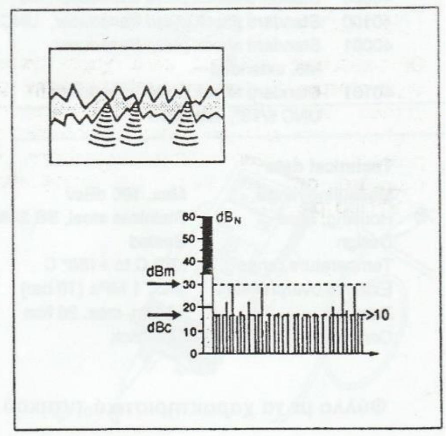
The rhythm of strong pulses is better found from the headphones, by setting the measuring level some dB lower than the $\boldsymbol{dB_M}$ level.



A characteristic of the bearing pulses is a random row of strong pulses (not discreet rhythm). Rhythmic shocks can emanate from a bearing but most of the time its appearance shows that an interference occurs. Subsequently, here are described some typical cases of shock pulses forms that may have more than one causes.

Measurement on a well condition bearing

A well condition bearing must have the $\boldsymbol{dB_M}$ value less than 20 and $\boldsymbol{dB_c}$ value about 5 to 10 dB lower. If a bearing gives such a measurement, we do not need to do anything else.



Maximum value may result lower than 0. Yet, in that case or when the value results too low, we should watch out the measurement because we may have selected a bad measuring point, or we may have misused the sensors. It is better to check the connections as well as the data. Is there no load to the bearing?

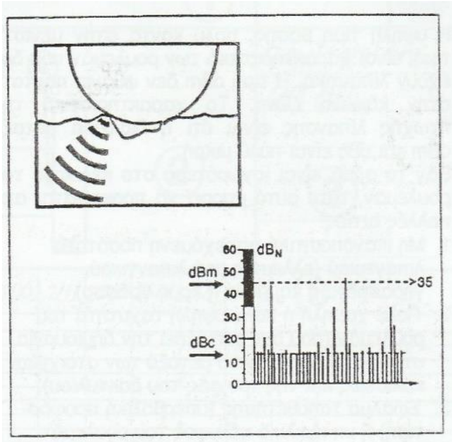
First signs of fault

Values of $\boldsymbol{dB_M}$ between 20 and 35 dB (yellow zone) and a medium increase of carpet value $\boldsymbol{dB_c}$ are a sign of tendency or slight fault occurrence on the bearing surface. We must note that in this case the difference between $\boldsymbol{dB_M}$ and $\boldsymbol{dB_c}$ increases.

Bearings that have $\boldsymbol{dB_M}$ values in the yellow area must be checked more frequently so to be specified if their condition is constant or is getting worse.

Signal from a faulty bearing

The form of measurement shown in the adjacent figure is typical for a faulty bearing, which revels: $\boldsymbol{dB_M}$ value greater than 35 dB, a large difference between $\boldsymbol{dB_M}$ and $\boldsymbol{dB_c}$ and a random form of strong shocks. The level of maximum $\boldsymbol{dB_M}$ value indicates the degree of failure:



35-40 $\boldsymbol{dB_N}$ indicates slight damage

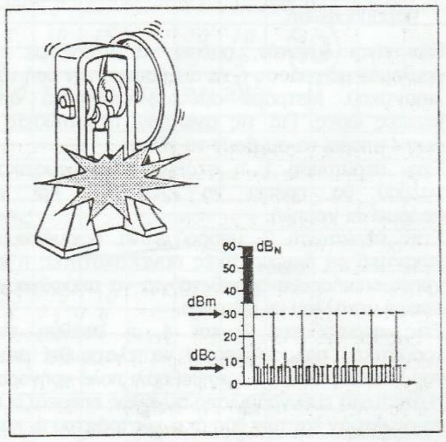
40-45 $\boldsymbol{dB_N}$ indicates serious damage

>45 $\boldsymbol{dB_N}$ a high risk of operation shut due to this damage

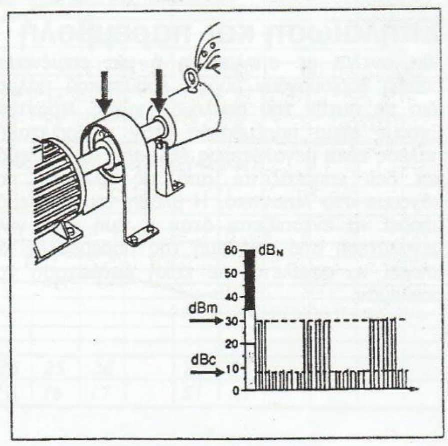
We must note that a similar view appears when contaminants occur in the lubricant (metal or other particles). The particles may come from the bearing itself, such as a surface peeling, or are transferred to the well condition bearing through the lubricant.

Rhythmic extreme values

The adjacent figure contains simple rhythmic extreme values. Simple extreme values may be caused by shock pulses or impulse pressures that develop during the normal operation of a machine. Other possible causes are the valve knock or loosen components that hit the machine hull at regular intervals.



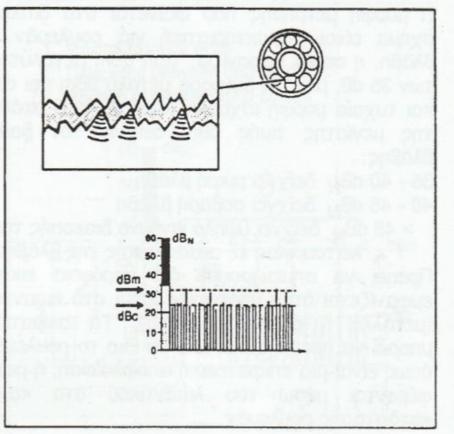
Periodic exacerbations

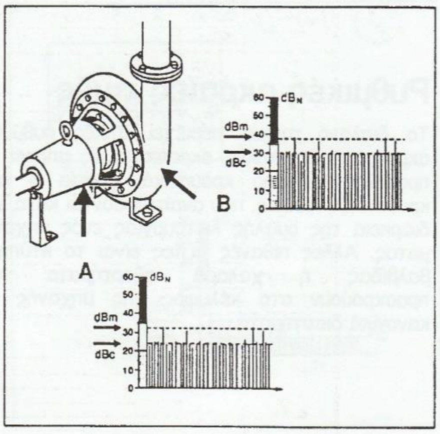


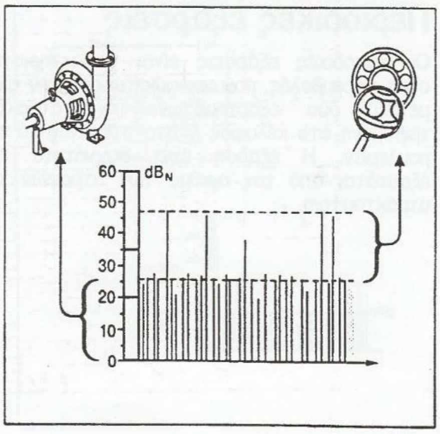
Periodic exacerbations are a typical interference signal, caused by the friction between two components, e.g., a shaft rubbing on the hull or on the sealing of the bearing. Exacerbation frequency depends on the number of speeds of the shaft rpm.

Samples of bearings with insufficient lubrication

The high carpet value, near the maximum value, is a characteristic of the bearings with no lubricant. $\boldsymbol{dB_M}$ Value will never reach the red zone. The characteristic of poor lubrication is that the difference between $\boldsymbol{dB_M}$ and $\boldsymbol{dB_c}$ is very small. If the signal is stronger at the hull of the bearing, then this can be caused by many reasons:







1. Unsatisfying provided amount of lubricant (insufficient lubricant flow, old or burnt or cold grease).
2. Very low or very high speed of the bearing (that does not allow the development of lubricant layer between the rolling elements and the ring's orbit).
3. Placement fault (excessive preload) or not cylindrical hull of the bearing.
4. Misalignment or shaft with bending deformation.

We must lubricate the bearing with grease or increase the lubricant flow, if possible. We measure directly and after a few hours. For the case 1 to 4 above, the following can happen:

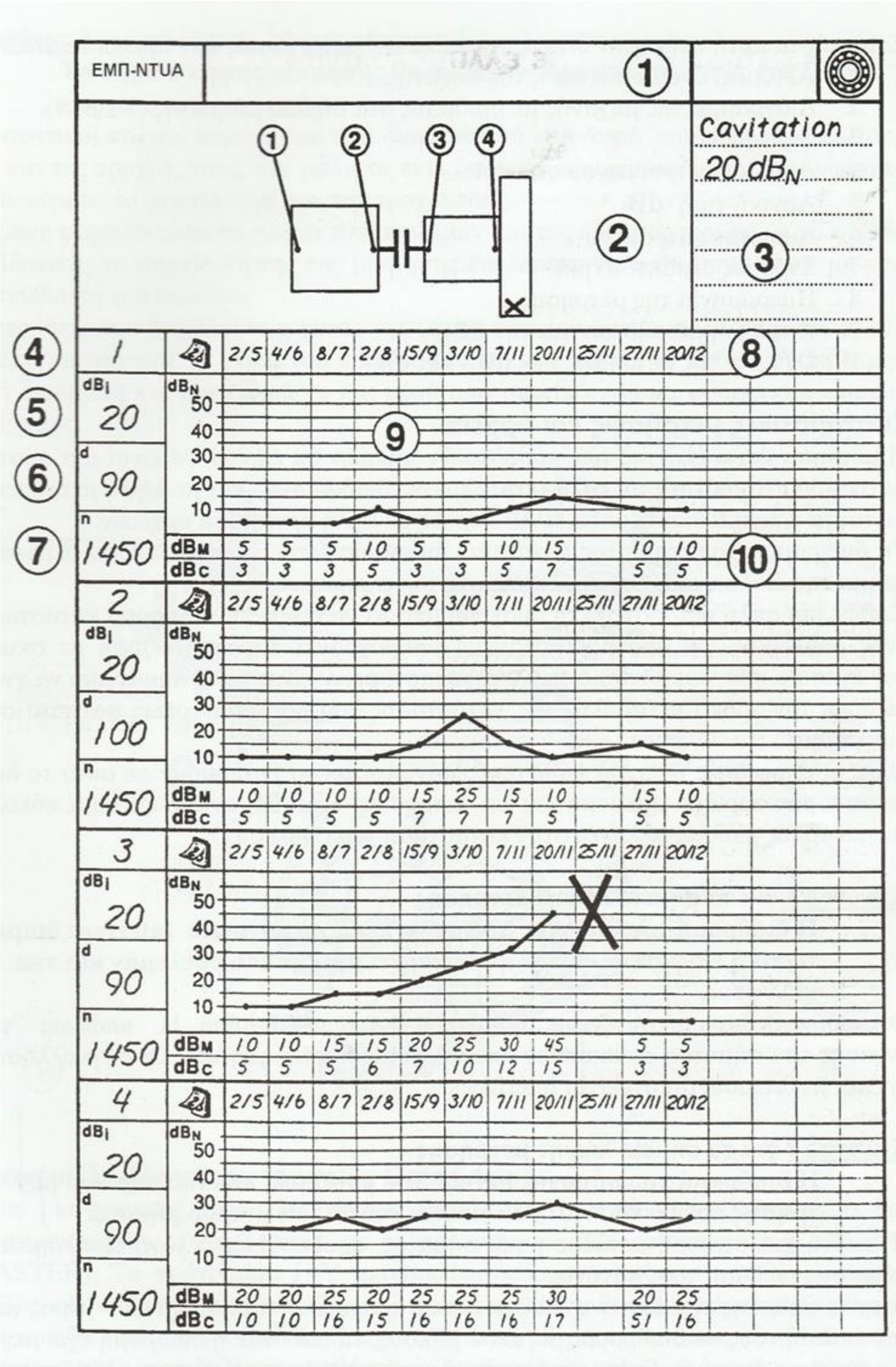
In case 1 the shock level must be reduced and remain low.

In case 2, we can try lubricants with various coherences, or use additives to avoid the metal-to-metal contact.

In cases 3 and 4, the shock pulse level can be decreased after lubrication, but it can increase again very quickly. The failure of alignment usually effects the bearings of both shaft edges and of both coupler sides too.

Cavitation and interference

A pump with cavitation or a persistent contact creates a same shock pulse form as this of the bearing with no lubricant. We have an interference signal when the shock pulse is greater outside the bearing and is not affected by the changes to the lubricant. The fault of the bearing can be detected when the $\boldsymbol{dB_M}$ value gets higher than the interference value and may be due to the bearing's poor condition.



Shock pulses monitoring card

There are note places at this card for the tasks that need to be done:

1. Number, name and position of the machine.
2. Machine chart with numbers at the measuring points (1, 2, 3, 4)
3. Notes.

4. Number of measured points.
5. Initial value $\boldsymbol{dB_i}$.
6. Shaft diameter.
7. Shaft speed rpm.
8. Date of measurement.
9. Measurements curve for $\boldsymbol{dB_M}$.
10. The values of $\boldsymbol{dB_M}$ and $\boldsymbol{dB_c}$ measurements.

Systematic measurements and files

The measurement and control of shock pulses must be a systematic task with well-organized procedures and the receipt of measurements at regular intervals, but mainly with proper file organization.

The chart of the figure is a follow-up form that comprises a main part of precision process in maintenance.

As a simple measurement with the device can indicate the current bearing position, for a scheduled long term monitoring with the aim of timely scheduling of the replacements, is very important to know how the bearing situation changes as shown at the chart.

All our decisions for each bearing, should be based on that chart which contains dates and measurements. So, we will easily recognize the sharp or other changes at bearing condition.

Rule 1 for the point of measurement receipt

The path of the signal must contain only one interface, this one that occurs between that bearing and the hull.

Any interruption of material continuity reduces the signal. The loss of signal intense to interface can be excessivelly big. The welds must be considered as interfaces.

Rule 2 for the point of measurement receipt

The path of signal between the bearing and the measurement point must be as straight and short as possible.

The correct and reliable signal transmission presupposes selection of the measurement points according to the rules.

The cause that rule 2 occurs is that the shock pulses lose part of their intense at long distance paths, especially when the path has a large cross section. Also, intense is reduced when the signals change direction of movement in the material.

<u>Rule 3</u> for the point of measurement receipt

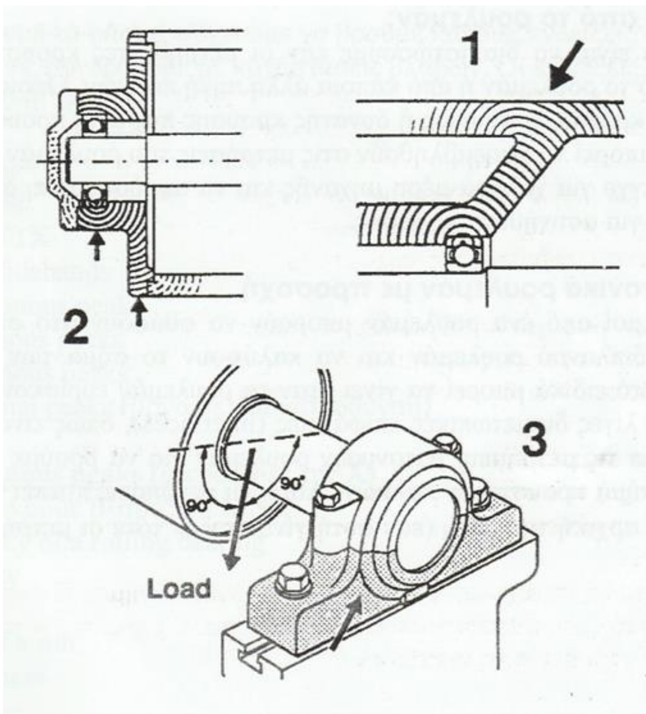
The point of measurement receipt should be in the load zone of the bearing.

Shock pulses are produced on the interface of the rolling bodies and their orbits, and even where the rolling bodies bare-carry their load, they touch their orbit.

The load zone is the part of the bearing and the bearing that carries the load.

The measurement receipt point should ideally be aligned with the direction of the load.

Measurements received out of the load zone give much lower indications because then the shock pulse signal should follow a twisted path and may also go through an interface to reach the measurement point.



Data logger device

When a device is a data logger it must always be used in combination with the computer software program (e.g., Condmaster). The computer program saves data from the measurement points as well as notices etc. It is used for creation of measuring commands that will be loaded to data logger. After the measurement, the results are transferred again to the computer for save, processing and evaluation monitoring, e.g., SPM, VIB, EVAM, SKF, FAG, etc.

General rule. We are obliged to evaluate measurements and shock pulses

- When the results have high values.
- When the measurements come from new measuring point receipt.

We must not decide to replace a bearing before we search and ascertain that:

- The measurement and the result are correct.
- The measured pulses actually come from the bearing.
- The cause for high measurement value is the fault of the bearing.

Does the signal come from the bearing?

The first step is to ascertain if the measured shock pulses come from the bearing or another pulse source. Any kind of metallic click, scratching or strong impact **produces** shock pulses that can interference to the measurements of the bearing.

Check for loosen machine parts and listen carefully to the machine for unusual sounds.

We measure neighbor bearing carefully

Impulses from a bearing can reach the measurement receipt point of an adjacent bearing and get above the signal of its shock pulses. This especially can be done when the bearings are on a machine hull with a few interfaces, such as transformers. We compare the measurements of the neighbor bearings to find the source with the stronger impulse signal. These measurements must be received with the same initial value $\boldsymbol{dB_i}$ (if this becomes zero then the measurements are absolute).

CHAPTER 10 FAULT DIAGNOSIS

10.1 MACHINE FAULTS AND IDENTIFICATION WITH FREQUENCIES

When we analyze the spectrum we ask for some indications and then we try to connect them with fault situations. The indications we ask for in the spectrum are:

High 1X peak

High 2X peak

Harmonics of Sidebands

Other synchronous peaks

Sub-synchronous peaks

Non-synchronous peaks

Very directional peaks (only one direction)

Sub-synchronous peaks (lower than X)

In these we find faults oil whirl,

Cage frequency of a rolling bearing

Belt frequency

Turbulence

Indications of a rub

Severe looseness

Synchronous peaks (integer multiples of X)

In these we find faults such as

Imbalance

Misalignment

Looseness

Bent shaft

Blade and vane wear

Gear mesh

Non-synchronous peaks (not integer multiples of X)

In these we find faults from:

Rolling element bearing

Component of other shaft

Harmonics of sub-synchronous frequencies

Resonances

Noise from other machines

Cavitation

Combustion

10.2 IMBALANCE

Imbalance is the situation during which the geometric axis of the shaft does not coincide with the inertia axis of its mass, or in other words the mass center of the shaft is not on its rotation axis. Scilicet there is something heavy during rotation and along the shaft instead of the rotation shaft (eccentric). Centrifugal force is developed on the eccentric mass, which strains the bearings (its value in a given direction varies during rotation due to the circular motion).

If we consider the example of the fan with the eccentric mass on its blade, the force is maximum only towards the radius that connects the center to the position of the eccentric mass. This force varies sinusoidally, in a constant point on the hull of the bearing. The magnitude of force is proportional to the rotating speed and the size of the eccentric mass.

If a shaft has imbalance we expect to see a sinus signal in the waveform with the same frequency as this of the shaft and a great peak to the spectrum on frequency 1X. But actually, there will also be other vibration sources (looseness, misalignment, wear on bearing, noise etc.), and therefore the signal is not clearly sinusoidal.

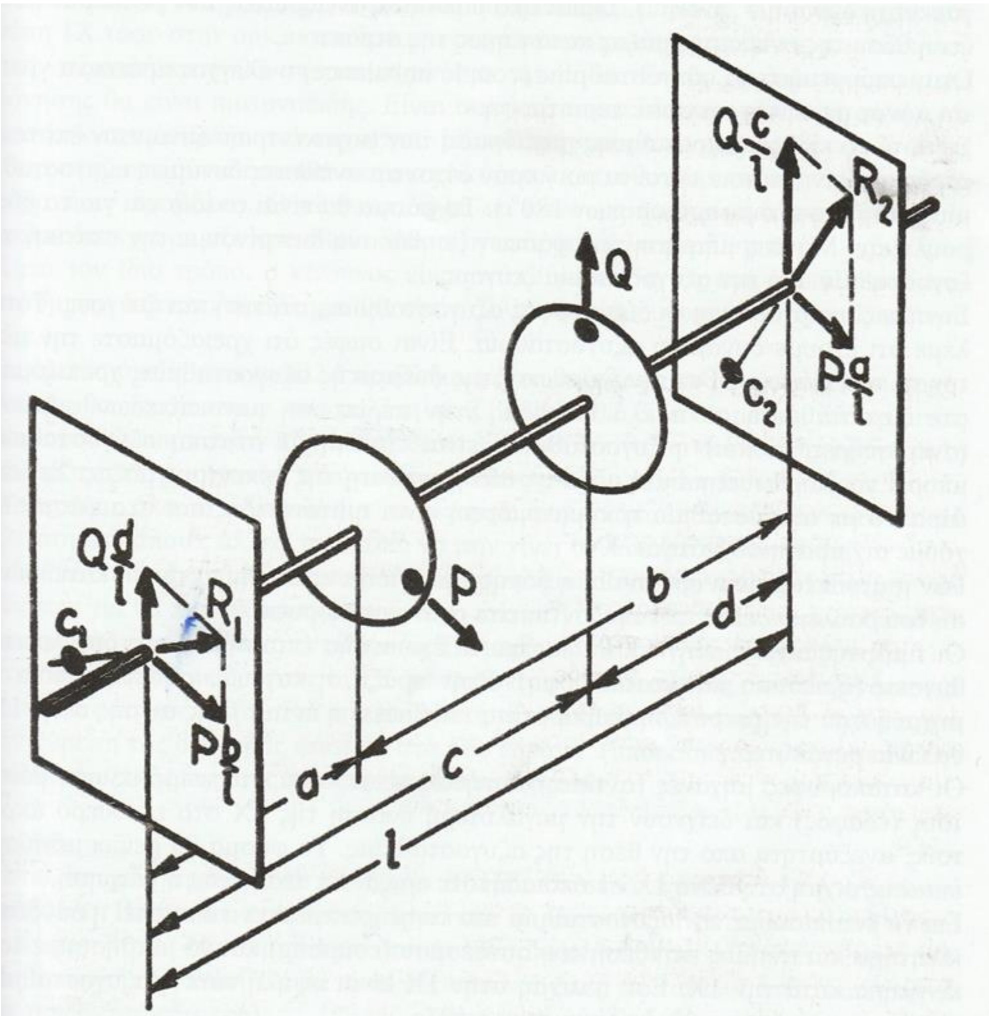
Every rotating machine component will have some remaining imbalance (no components with absolute balance occur). The result is that the peak of the eccentric masses will always exist at frequency 1X, and if the rest of the machine has no other vibration causes then its peak 1X will be dominant to the spectrum, as the waveform is sinusoidal.

Depending on the peak level we have to decide if the imbalance is dangerous or not.

When we say that there is a state of imbalance?

The criterion is the vibration magnitude at 1X position, which cannot be independent of the machine's size and the rotation speed.

Generally for these limits the Standard tables apply.



Sketch 10.1 Dynamic balancing requires correction on two levels.

There are two ways of imbalance, the static and the couple.

The simple type of imbalance is the static form that equals to an eccentric mass in a position of the shaft. It is called static because it is seen even when the shaft does not rotate (rotates the shaft so that the eccentric mass to go in the lower position).

Static imbalance effects to 1X on both bearings in such a way so the reactions are always towards the same direction (they are in phase).

Static imbalance will give a high peak to 1X frequency of the spectrum. The peak magnitude is proportional to the imbalance (eccentric mass and eccentricity) and of the square of the (angle) rotation speed (centrifugal force $= m \cdot r \cdot \omega^2$). An important play to the bearing reactions plays the position of the eccentric mass along the shaft.

When a couple imbalance exists the control must be done only with the shaft rotating.

In that type of imbalance the effect of the centrifugal forces on the bearings is such so the bearings receive opposite imbalance forces (that have a phase difference of 180º). The spectrum will be the same for both bearings. Only the phase measurement assists to separate the static imbalance from the couple imbalance.

Usually both of the imbalance types, static and couple occur. Then we say we have a dynamic imbalance. It is clear that we need the phase measurement. We need balance in many levels to correct the dynamic imbalance. In the case of single-level carriers (fans, plates) balancing is easy. Static imbalance can be fixed with a mass at the opposite position of the eccentric mass. The waveform is sinusoidal and there is a peak to the spectrum in frequency 1X in a shaft with imbalance.

If the shaft is horizontal, the vibration measured across the radial direction of the bearing, either it is in the horizontal or the perpendicular axis.

The peak values in 1X should have the same intension on both directions (horizontal and vertical). Actually, the direction in which the machine will the peak intense have the smaller stiffness at 1X will be the larger.

Perpendicular machines (pumps) are usually well fixed on their carpet (ground) and show the higher intense of 1X at their free edge regardless of the imbalance position. The spectrum will show a high intense peak at position 1X at any horizontal position the measurement is done.

The connection between the pump and the engine must be interrupted in the coupling position in order to detect the imbalance of the engine and measure the engine by 1X. If 1X peak is high then the imbalance is in the engine, otherwise is at the pump.

Pumps, fans and pulleys in cantilever are a usual phenomenon on industry. Shaft in cantilever = Overhung rotor = cantilevered rotor.

On cantilevered machines the peak on 1X position is important to all directions (horizontal, vertical, axial). The measurement must usually be done on the bearing that is near the cantilever (question: does it matter if the bearing is of fixed or free bearing?)

Due to the bending tendency that develop by the cantilevered mass on the fixed bearing, the bearing transfers the bend to the hull with a couple of axial forces, through which the hull is axial vibrated.

There are various causes for machine imbalance. One of the most basic is to be imbalanced from the beginning (lack of education, lack of balancing devices, lack of time).

In the case of imbalance occurrence we expect: a high intense peak in position 1X to the horizontal and the vertical direction and furthermore to the axial direction for cantilevered machines. Also the waveform of the vibration will be sinusoidal. The results and the analysis are clear. We recognize the speed 1X and search for vibration measurements to the horizontal and the vertical position (and in axial for cantilevered machine). The risk is to diagnose wrongly, that is to say that there is imbalance when another problem already occurs.

By the same way, the risk is to tell that a machine need to be balanced, when the problem occurs somewhere else (poor balancing, looseness etc.). We firstly organize the other problems and we finally go to balancing.

As we go ahead the fault study we will also find other reasons that cause high peak in position 1X, such as misalignment, machine looseness, the bended shaft and the eccentric engine. We read all data to ensure some causes and reject some others, in order not to occur a fault to diagnosis.

We should not forget that we can also use measurement of the phases to decide. Both of the bearings bear vibrations in phase at static imbalance (1X are measured in the same direction). In couple imbalance the 1X are measured in different directions (with a phase difference of 180º). For the dynamic imbalance there is no rule to find the phase difference at the two bearings (we can search with the total intense device to the circumference of the bearings).

<u>Example</u>

Even that a machine has a 20 HP engine, elastic joint and a cantilevered pump. The pump has 6 blades and supplies 105 lit/min with pressure (head pressure) 34

m. remember that if a cantilevered machine has imbalance, the peak at 1X position will be present to all directions (vertical, horizontal, axial).

The measurements have been done to the bearing that is towards the cantilever as following:

In vertical direction the vibration is sinusoidal to the waveform while at the spectrum has a peak in position 1X. The deviations of smooth sinusoidal signal are due to the signals of frequencies 2X (from the joint) and 6X (from the pump's blades).

In horizontal direction the vibration to the spectrum has the fundamental peak in 1X position, and even has more intense.

Finally, in the axial direction the vibration at spectrum has the fundamental peak and even of high intense.

Here we have a cantilevered machine with a basic imbalance problem.

The measurements at the other bearing gave lower vibration peaks.

There is **eccentricity** in a component from the rotating center when the geometrical center of the pulley, the cogwheel or the bearing etc., is in some distance from the rotation axis.

The symptoms of eccentricity are the same as in the imbalance.

Eccentric cogwheels create high peaks at the radial direction of position 1X.

Eccentric belt pulleys create high peaks in position 1X in a parallel direction with the belts.

There will be a high peak in belt movements in position 1X on both shafts (drive and moving pulley).

Two peak will appear (one for each shaft).

At belt movements we control the eccentricity of the pulleys by removing the belt and by checking the 1X peak on each one's axis.

(Question: how are we going to rotate the axis of the second pulley when the shaft will be removed?)

10.3 MISALIGNMENT

When the shaft axis that connect to the coupling do not coincide we have a misalignment. Parallel (offset) misalignment = when the axis are parallel in some distance.

Angular misalignment = when the axis intersect (under some angle).

In practice all misalignments are due to the combination of two.

Angular misalignment causes bending tendency in each shaft. This tendency also creates at the other two bearings that are near the coupling, to the axial direction, peak of high intense in position 1X and of low intense in position 2X.

In radial directions (horizontal and vertical) low peaks will develop in positions 1x and 2X.

Parallel misalignment causes bending tendency and coincident force at the edge of each shaft. This tendency creates problems to both bearings.

In radial directions (horizontal and vertical), in each of the bearings near the coupling, peaks are developed in positions 1X and 2X. The peak in 2X is usually higher than the peak in 1X.

The peaks in axial direction in positions 1X and 2X are low in shafts with parallel axis.

On the other side of the coupling vibration have a phase difference of 180º to the axial and the radius directions.

Generally, the diagnose is based on the increased vibration intense at peak 2X and at the increased peak in 1X along the axial direction but also in one of the two axial (horizontal or vertical) directions.

The elastic couplings add harmonic 1X and 2X.

Misalignment (lack of alignment) creates various problems to various machines. Each case must carefully examined. Beyond the peaks in positions 1x and 2X, sometimes a peak also appears in position 3X.

It is a way to distinguish if something comes from misalignment or imbalance and increase the machine velocity.

Vibration due to imbalance changes with the square of velocity, while vibration does not change due to misalignment.

Another way of controlling is to isolate the shaft connection and operate the engine. If the peak in 1X continuous to occur, then there is a problem with balancing of the engine. If peak 1X disappears, then either the second shaft (or its components) have the problem of imbalance or a problem of shaft misalignment occurred.

We ask for the solutions.

Machine alignment will happen in some temperature, yet due to the thermal expansion and contraction, let us hope that it will cover the operation temperature.

Vibration measurements for alignment should be done when the machine is at operation temperature.

Axis misalignment is due to the following basic causes:

- Poor component assembly (engines, pumps, pulleys, couplings, wheels, etc.).
- Relevant displacement of the components (geometry change) after assembly.
- Deformation due to forces and tendency due to pipes, couplers, bases, etc.).
- Deformations due to temperature differences.
- Poor placement of flanges and couplings.
- Soft bearings and poor connections of base bolts.

A main cause of misalignment is that alignment did not occur from the beginning, either due to lack of trained stuff, or due to lack of devices, or due to lack of time.

Because the presence of misalignment is accompanied in the spectrum by the peaks 1X, 2X and 3X we need to precisely calculate the rotating speed of the machine.

The analysis must be done to the data of all three directions (horizontal, vertical, axial). The measurements along axial direction play a significant role for misalignment maintenance.

In misalignment vibrations are not equal along the vertical and horizontal direction, possibly the one is two times the other.

The collection of independent measurements to all three dimensions is difficult (many times there is no space), unless we use a three-axis accelerometer.

Some times 2X (100 Hz) is identified with the harmonic of the current (50 Hz).

But 2X is an indication of misalignment.

The waveform many times does not help. If we have a strong peak in 1X to the spectrum of axial direction we would expect to have a sinusoidal form to the waveform. But if we have a strong peak in 2X and some lower peaks to the spectrum, then the waveform has typical form.

Example

A machine has electric engine AC 20 HP and moves a centrifugal pump with 6 blades through an elastic coupling.

Measurement to pump – in **vertical** direction: we have peak to the spectrum in 1X but also in 2X, peak 2X is to be explored.

Measurement to pump – in **horizontal** direction: we have peaks to the spectrum in 1 and 2X. We have harmonics of 2X.

Measurement to pump – in **axial** direction: we have peaks to the spectrum in 1X and 2X. The intense of 1X is lower hear than the corresponding values in the vertical and horizontal direction.

The peak in 2X assures as for the occurrence of misalignment.

The occurrence of bent shaft is similar to the imbalance and the misalignment. A bent shaft mainly develops high peak 1X, in axial direction, when the bend of the axis is near the middle distance of the bearings. If a bend is closer to the coupling, then a 2X peak is developed. Horizontal and vertical (axial) directions show peaks in positions 1X and 2X, to the bent shaft, yet the dominant vibration measurement is this of the axial direction.

10.4 LOOSENESS

The condition called looseness creates important peaks in position 1X. There is the Rotating looseness or non-rotating looseness.

Rotating looseness is caused by a large grace between rotating and fixed components of the machine (such as to the rolling bearings).

Non-rotating looseness is created between two, under normal conditions, still components, such as the legs of a machine and the foundation, or the outer ring of the bearings and the machine hull.

Rotating looseness is due to the wear of the bearings.

Large axial grace to journal (sleeve) bearings and bearings produces 1X harmonics that may exceed even the 10X.

As the looseness condition of the machine gets worse, peaks increase to intensity and degree. Some peaks that will coincide with coordination frequencies of the construction or with vibration sources (such as the blade frequency) will become higher than the peaks in other frequencies. Great grace to journal bearings may cause harmonics of 5X that are called half-order harmonics, which can be produced from friction and impulses (transient).

Even harmonics of X/3 can occur.

Example

This example refers to machine measurements with great looseness in the bearing foundation.

Harmonics in vertical direction are due to non-rotating looseness (foundation flexibility).

Harmonics in horizontal direction have an intense three times the vertical direction.

Frequently transients are visible in the waveform.

Although there are harmonics, the peaks are too low in axial direction.

The looseness at the support system of the machine will increase the vibration in position 1X in the direction where the lower stiffness occurs. This direction is usually the horizontal.

On vertical machines is difficult to distinguish if the vibrations comes from looseness or elasticity. In case of looseness the horizontal 1X is larger than the vertical 1X.

In case of imbalance horizontal 1X is smaller or equal to vertical 1X.

If a machine has elastic foundations, than the vibration will always be greater than the horizontal direction. Looseness (elasticity) of foundations can come from loosen coupling bolts and from wear or coupling breakage.

There is a phase difference of 180º between the vibration, the machine and the foundation measurements, in vertical direction.

The spectrum will show peaks at frequencies 1X, 2X and at most 3X to a loosen foundation of the bearing.

In intense cases 0.5X will also appear.

There is a phase difference of 180º between the vibration measurements on the bearing and the foundation.

Example

Electric engine moves a centrifugal pump with 4 blades through an elastic coupling, if we consider the measurements in the vertical direction we see the in the spectrum harmonics of 1X. We also have harmonics of 0.5X which indicate that the looseness is great.

10.5 FAULT DIAGNOSIS TO A BEARING

Bearings occur in all rotating machines and are responsible for many faults. Yet, the bearing is designed to operate for many years if it is placed and maintained properly. Under 10% of the bearings will operate until the calculated lifespan, because

About 40% fails due to poor lubrication

30% fails due to poor placement

20% fails due to overload causes etc.

If machines had followed the precise maintenance rules when it comes for balancing and alignment and if they weren't operating around the coordination frequencies and if the bearing were properly lubricated, then they would be more reliable and operate until their determined lifespan without faults.

Unfortunately, these do not happen and the bearing fail very early. The maintainer's task is to detect the early stages of the upcoming faults and estimate the seriousness of the problem.

The vibration analysis, the pulse measurement and the analysis of the wear particles in the lubricant are very good tools for that purpose.

A bearing with fault produces to the spectrum peaks to frequencies that are non-synchronous (are not integer multiples of 1X).

The occurrence of non-synchronous frequencies to the spectrum indicates the possible existence of problems to the bearings. In these cases we look for the frequencies of the bearing which have been calculated in another point.

BPFI = BPI·f = ball pass frequency inner race – inner ring.

BPFO = BPO·f = ball pass frequency outer race - outer ring.

FTF = FT·f = fundamental train frequency – cage rate – cage frequency.

BSF = BS·f = ball span frequency – rolling body frequency.

Where f the frequency of the shaft. The forms of frequency calculation of the bearings apply for the ball bearing and the other rolling bearings. From the rest forms of calculation results that

BPI + BPO = N = number of rolling bodies

E.g., if BPI = 4.8 and BPO = 3.2 then N = 8.

The inner bearing ring is a body with the shaft and will appear 1X sidebands. By these we can find BPI and we can calculate BPO from the number of the rolling bodies.

For a bearing with number of rolling bodies N apply the relations

$BPI = 0.6 \cdot N \quad BPO = 0.4 \cdot N \quad FT = 0.4$

Provided that we can find these frequencies to the spectrum, we will now look into the way of bearing destroy in nine steps.

Step 1.

During the first operation stage of the bearing the frequencies that are due to fault indication signals are very high of 20 Hz up to 60 kHz or higher. There are devices which detect these frequencies, such as the Spike energy, HDF, Shock pulse, SEE and possibly also to other devices that measure Ultrasonic frequencies. The normal vibration spectrum in that phase will not show information about the bearing.

Step 2.

Now faults are quite large to cause ringing (like the bell) to the bearing, with the physical frequency of its coordination. This frequency also reacts as a configuration carrier of the bearing’s fault signal (modulation, as happens with the melodies of the radio station).

Step 3.

Now faults have progressed and the bearing components’ frequencies will begin to appear. Let’s consider that we have a fault to the inner ring of the bearing so appearance of a peak to BPI.

As the wear and the fault gets bigger, the vibration intense (the size of the peak) to BPI frequency will also get bigger.

Here must mentioned that many times the peaks to the frequencies of the bearing sometimes appear and sometimes disappear. That means the question is, does the fault occurs or not?

Step 4.

As the wear gets larger and the fault gets worse, the bearing frequencies will develop harmonics, which means that shock phenomena begin to appear. Sometimes peaks begin to appear in the non-synchronous frequencies before the peak to the main frequency 1X is developed.

The height of the peaks to the bearing frequencies is low at the beginning. That's why we should avoid the linear axis scales and use the logarithmic scales.

Some argue that we must not concern for a signal that cannot give a peak to the linear scale spectrum.

The proper is that when we plan to take the warning signals of the upcoming faults on time we should use anything that can help us (as the logarithmic scales to the spectrum) achieve our purpose.

In a logarithmic scale the peak in 3.18X is shown while in the linear scale is not shown (in a taken measurement).

The waveform starts to be interesting.

Impacts appear to the waveform as pulses of releasing energy. We can see peaks of high frequency but of small amplitude, which are superimposed on the waveform.

In order to discover the early stages of fault on the bearing, we should collect a high frequency waveform in acceleration values with an accelerometer without the integration of the measurements.

Step 5.

As the wear gets larger and the fault gets worse, the vibrations get greater and more harmonics appear. Now the sidebands begin to show to the spectrum depending on the nature of the problem.

We must remember that modulation happens when the oscillation amplitude of a frequency (here let us consider that is the frequency of the inner ring of the bearing) changes periodically.

Example

We will examine the case of modulation occurrence to a horizontal machine. If the inner ring of the bearing has wear (e.g., a puddle) the impulses developed (when the rolling bodies hit on it) will be more violent (will give high peaks to the spectrum) in case that the wear is found in the load zone of the bearing.

When the wear is found out of the load zone the intense gets lower and will become minimum when the wear is found 180^{o} opposite the center of the load zone.

In that case, BPI (inner race defect frequency) of the inner ring is modulated by the rotation frequency of the inner ring (which is identified to 1X of the shaft with which is a body), i.e., around BPI we are going to have sidebands in distances 1X.

If the rolling body has an issue, we will have again signal modulation for the same reasons. When the rolling body is at the load area, the impulse phenomenon will be more violent than in the case of the rolling body's position out of the load zone. The closer the rolling body to the load zone is the higher is the peak to the spectrum.

Rolling bodies move around an inner ring with their cage, to wiz with FT frequency (train-cage frequency), which is lower than 1X (about 0.4X).

When we start to notice harmonics especially when the sidebands begin to appear, the wear to the bearing is already visible. Now is a proper time for changing the bearing.

Note that in that stage, the high frequencies measurements such as spike energy and shock pulses have an upward trend. The waveform will show the effects of the fault to the bearing. The peaks to the waveform are going to be even higher and we will be able to measure the time between two peaks (occurrences) to calculate the frequency of the phenomenon that cause them.

Step 6.

The peak to frequency 1X will be amplified further and harmonics of 1X will appear, that are due to the fact that we have an increased grace, as far as the great wear increased the grace (clearance).

Step 7.

Now the peaks to the frequencies appear to the spectrum to be based on curvatures, which come from the noise of the coordination ringing of the various components to their physical frequencies.

We might can hear the bearing’s noise. The vibration values of the high frequencies probably they will begin to decrease. If you measure with such devices and see decreased measurement values, don’t think that everything is fine. Begin to order bearings and prepare the machine and the technicians for interference.

Step 8.

Now things are getting worse. The harmonics are getting greater due to the increased grace from wear and to the looseness development. The noise curvatures to the waveform are getting bigger and the carpet noise will start to raise, although the measurements of high frequencies are decreased. The bearing noise is definitely heard. It is too late for precision actions.

Step 9.

Do not expect to see something to the spectrum in this phase.

You received so many warnings that the machine did not operate well.

Woe! It has stopped operating.

Example

A machine has an engine of 40 PS and moves a blade of 16 blades through belt movement 2:1. We will watch the measurements done as time passed. The measurements to the horizontal and the vertical direction did not show something important at the beginning. The peak heights were low. The waveform shown effect by high frequency phenomena, i.e., there were some early stages of wear to the bearings.

Due to belt moving there were peaks from the belt flow and the shaft of the fan.

The measurements to the axial direction presented some interest. We were noticing a peak in 1X, which was probably due to misalignment of the pulleys (which probably created problems to the bearings).

The height of the peak got higher to the next measurements and shown more signs of wear to the bearing. In a logarithmic scale we noticed the peak to frequency 3.1X with harmonics. We are in step 4.

The steps of the bearing became very strong in the measurements that followed. The waveform gave information. We were seeing impulses (events) that happened with a time difference that corresponds to frequency 3.1X.

The precise value of the frequency above was 3.06X.

In the spectrum with an opening of 100X we saw its harmonics and how strong they were.

We said that we will have sidebands to the bearing.

a. With a fault to the rolling body that has forcing frequency BS and sidebands of FT.

b. With fault to the inner ring that has forcing frequency BS and sidebands of 1X.

We didn’t have sidebands to the spectrum, which means that there was problem neither to the rolling bodies nor to the inner ring.

The measurements referred to the vertical direction were done after a long time. The waveform livened up. The peak got higher. Noise curvatures appeared and the carpet noise got bigger. Everything seemed different and out of logic at the last measurements that were taken. The waveform did not have the periodic sharp peaks of the impacts. The spectrum did not contain harmonics and the carpet noise was very big. We are in stage 8. The bearing was almost ready to destroy. This happened a few hours later.

At the first steps of the bearing wear, we notice vibrations of the bearing with its physical frequency, as the bearing rings with its frequency.

The component of the bearing that has the fault bears the impulses and is forced to ring causing sidebands to its defect frequency. E.g., if an inner ring has wear then in its defect frequency BPI modulation will be created and sidebands will appear in a distance one from the other equal to the rotation frequency of the inner ring which is the 1X. As is internationally said:

The bearing is ringing = the bearing is resonating with the defect frequency sidebands. Demodulation of a signal = applying filters to remove frequencies.

The initial signal will have many vibration sources which will have peaks heights higher than these that the coordination peaks of the bearing have.

In the waveform we can see the basic sinusoid signal with lots of shock signals on it.

The transient phenomena are due to ringing of the bearing.

We remove all the frequencies that are lower than 2000 Hz to analyze the bearings and with a High pass filter we allow the high frequencies to remain to the spectrum, so there are only high frequencies to the spectrum and the waveform contains only impulse loas that give us information.

We fold over the higher sidebands to the beginning by using a demodulator that reacts as a rectifier, that is makes the negative values positive.

This is the result of signal rectification.

The signal rectification turns the negative values into positive.

By using a Low pass filter we stay only with the sidebands.

The purpose is only to remain the signal from the modulation source we are interested in. the frequency range (Fmax frequency range) that we check is usually 15X-20X.

The final signal must contain a strong SET of harmonics. The basic frequency is the frequency (defected frequency) e.g., if the fault is to the inner ring the basic frequency is BPI.

The measurements will contain lots of information and the spectrums will be complicated in practice. We do not measure the heights of the peaks (as in normal spectrums) in demodulated spectrums. We compare peak heights to see how the fault is developed. When a significant fault occurs, the peaks are much higher (20dB or 100X) from the noise floor.

As the fault gets bigger and we enter step 7 or step 8 the noise floor gets louder and approaches the peak height.

This is a really bad situation.

The demodulation process above can be also applied to other fault areas, such as: Gear mesh analysis, motor current analysis, air gap eccentricity analysis of electric motors etc. modulation sources.

We can detect which bearing has a fault by applying this method.

This method can also be applied to Gear mesh.

While the wheels cooperate, the teeth to teeth contact creates a vibration of high frequency likely Gear mesh frequency. This frequency is a carrier. A damaged teeth that belongs to the one wheel causes shock pulse as it passes through the Gear mesh and brings on amplitude modulation around the Gear mesh frequency, as we will see below.

10.6 FAULT DIAGNOSIS TO COGWHEELS

We know the main about frequencies (forcing frequencies) of a one-level cogwheel transfer, as is the frequency of the input shaft, the frequency of output shaft and the frequency of the Gear mesh that is equal to the number of teeth of a wheel to the frequency of its shaft. We have given examples for multilevel motion transfers with cogwheels. We will find peaks to forcing frequencies such as:

1X = to the input shaft at the mechanism.

1X pinion 3 = to the shaft of the drive engine (here the input shaft is different than the shaft of pinion 3).

GM = Gear mesh = at the gear.

We may also find a peak to 2X, and probably find sidebands around GM.

The peaks will be noticeable in these frequencies in the axial directions of the cogwheels with straight teeth and in axial direction for helical teeth.

The waveform is of great importance at cogwheel boxes. We notice a pulse to the waveform in each tooth contact.

We see the pulses clearly. Each pulse refers to the contact of two teeth (one of each wheel). We also distinguish the circle related to a complete rotation of the shaft.

Let’s consider two cogwheels with alignment problem.

When the wear to the teeth begins the following will take place:

a. The increase of peaks to the sidebands will start (around GM frequency) that correspond to the wheel speed whose teeth are worn.

b. The stimulation of the physical frequency will begin (coordination) of the wheel. This peak will be accomplished by sidebands.

The peak height to the gearmash frequency depends on wheel load and the size of misalignment.

Therefore, in wheels with a great load the high peak at this position is not necessarily a problem.

A cogwheel with eccentricity or a cogwheel in a bent axis will give a sideband (at distance equal to the frequency of its shaft) near the frequency of the gear mesh.

A cogwheel with misalignment will give at frequency GM of the gear mesh and peaks to the sidebands based on the frequency of its shaft.

It will also give harmonics 2 GM and 3 GM with sidebands.

The spectrum must cover a large area in order these high frequencies to be included to the spectrum (frequency range F_{max}).

In a test with a misaligned wheel, a waveform with uneven wear to the teeth of the wheel (in a rotation). As we continue the TEST analysis and ZOOM at the harmonics of GM frequency we were able to see the 1X sidebands in that frequency to the measurement done in the vertical direction.

As we continue the TEST analysis and zoom at the GM frequency we can see the 1X sidebands to the harmonic 2 GM with a measurement done in the axial direction.

A wheel with crack or a breakage creates a high peak to the rotating frequency of its shaft and causes a stimulation of coordination of the wheel with its physical frequency, around which will be developed sidebands with the rotating frequency of its shaft.

The best way for someone to distinguish if there is any wheel with crack or breakage is through the waveform. In a cogwheel with 12 teeth, there must be 12 pulses at each shaft rotation one of them will be very different from the others. The time difference between the large pulses (events) corresponds to a rotation of the shaft or the wheel (each tooth contact once per rotation of the wheel).

The **coincidence frequency** of the same teeth from each wheel (hunting tooth frequency) is the frequency during which a specific tooth of the one wheel will cooperate with a specific tooth of the other wheel.

If the transmission relation is 3.0 (the number of teeth of the big wheel is three times the number of teeth of the small one), this frequency is equal to the frequency of the big wheel. We are going to have a teeth coincidence in each rotation of the big wheel. The coincidence frequency of teeth must be as smaller as possible.

In a pair of cogwheels, the coincidence frequency is found by the gear mesh GM frequency when we divide with the minimum common multiple of the number of the teeth of the two wheels. When the numbers of teeth are first against others (e.g., 19, 39) the minimum common multiple is equal to their product. In practice, the coincidence frequency of the wheels is used in order to detect the faults of the step wheels, which may have occur during the construction or due to abusing. This is a small frequency (much lower than 1X) and the vibration sounds very loud in the cogwheel box (as an angry moan).

10.7 FAULT DIAGNOSIS IN BELT MOVEMENT

Belt movements are a common and cheap way of motion transmission and power transfer. Yet, they often appear many problems such as: wear, misalignment, unsuitable tensile strength of belts and belt coordination.

The relation $\mathrm{d}_1 \cdot \mathrm{n}_1 = \mathrm{d}_2 \cdot \mathrm{n}_2$ is not absolutely true due to the sliding of the belts on the pulleys.

In fact it is used and gives quite precision.

As the wear of the belt movement gets bigger and the belt loosens, the previous relation changes as the length of the belt also does.

The first forcing frequency is the frequency BR of the belt (belt frequency or belt rate or fundamental belt pass frequency). BR is the frequency with which a point of the belt passes by a specific constant point of reference and is always lower than each pulley's frequency.

It is calculated by the relation $\mathbf{BR} = \boldsymbol{\pi} \cdot \boldsymbol{d_1} \cdot \boldsymbol{n_1} / \mathbf{L}$

Where L the length of the belt to its nominal dimension.

There is also a second frequency TBR for the timing belts that is the timing belt rate and is calculated by the relation

TBT = BR · Z where Z = the number of teeth of wheel 1.

If the belt is worn or loosened in a belt movement of two pulleys, peaks to the frequency BR and the harmonics of 2 BR and 3 BR, 4 BR etc. will appear.

The peak will be higher in 2 BR

At the test that was done the belt had bearded a fault at a point. Each time that the fault knocked the pulley a vibration was developing and a pulse to the waveform was noticed. The pulses (events) were visible to the waveform as the difference between them was giving the belt frequency. The transient phenomenon that is caused by this pulse creates lots of harmonics to the spectrum.

A **pulley with eccentricity** will give a strong peak in frequency 1X in the axial direction that is parallel to the fields of the belt or to the component of the belt forces. The eccentricity of one pulley at the belt-pulley system, that is shown to peak 1X, will also be shown to the 1X of the other pulley.

If we remove the belt, only the diseased pulley will give a 1X peak. Misalignment of the pulleys causes high vibration in 1X mainly in the axial direction. Harmonics of BR of the belt may sometimes develop in the axial direction.

When the coordination frequency of the belt (physical frequency) coincides with one of the two pulleys frequency (drive or driven) than we have a coordination to the belt. The coordination frequency of the belt changes either with the change of its tendency strength (by changing the distance of the centers) or with the placement of a new belt.

We can determine the coordination frequency of the belt, without operate the belt movement. The belt moves as a guitar string.

<u>Example</u>

By the hit on the belt (as a guitar string) is received the waveform and the spectrum shows a peak in one of the physical frequencies of belt coordination. Other physical coordination frequencies of the belt are also shown to the spectrum and the logarithmic scale.

Couplings

There are many types of shaft couplings. Their problems usually cause the same symptoms as these of misalignment.

These problems produce strong vibrations in 1X. If the coupling flanges are not parallel, the result is the same as this of the angular misalignment.

The coupling imbalance leads to high peaks in frequencies 1X and 2X in the axial direction. The worn couplings cause all the symptoms of misalignment and looseness.

10.8 FAULT DIAGNOSIS IN PUMPS, FANS AND SIMILAR

There are many types of pumps and the produced vibrations vary. The pressure to section and depression, the occurrence of air and the cavity affect the form of vibration. The centrifugal pump needs attention to the vane pass frequency. If the peak importantly increases means that there is a vane corrosion issue or a flow issue or a possible alignment issue. Many times harmonics of the vane pass frequency appear too.

Gear pumps are often used to lubrication systems and almost always have a strong vibration to the gear mesh frequency, which is the number of teeth of the one wheel to the shaft speed.

Similarly, other pumps create peak to the spectrum at frequency PV that is the product of the number of rotating parts (vanes, teeth, threads etc.) to the speed of its shaft.

When the peak intense increases and the harmonics of PV and sidebands appear, we must know that a problem of wear or flow exists.

Most fans are of axial propeller type or centrifugal. The fans usually address the issue of dust furring on their vanes. This creates imbalance problem and must be corrected directly. The fans are usually found in cantilever and so they show increased peak to 1X in the axial direction. If one of the vanes is damaged, the blade pass frequency will show high peak. If there is also a problem to other vanes sidebands to BP will appear with a difference of 1X. If there is a gap issue with the housing, harmonics of BP will develop. There is a problem to centrifugal fans with the unequal distribution of the speed to the input that creates increased vibrations to the frequency BP blade pass.

If the fan is imbalanced and in cantilever, a high peak will be developed to 1X in the axial and the radial directions. Vanes with faults create 1X-sidebands around BP.

It is good for a TEST with completely open dampers of the pipes to be done, so that the air flow not to be disturbed.

The centrifugal compressors have a spectrum similar to that of the centrifugal fan. Vane pass frequency CV is calculated by the number of the vanes and the speed of the shaft. Destroyed or worn vanes cause increased vibration peaks to CV and usually produce 1X-sidebands around CV. The air depression in the compressor output is a problem of fluid dynamics. It causes vibration at lower frequencies than 1X (sub-synchronous) due to inappropriate pressure in the output.

The most famous reciprocating engines are the piston pumps, the piston compressors and the internal combustion engines. The piston moves twice (up-down) in a turn of the shaft, i.e., we will have a peak to 2 in a problematic piston. At a 4-time internal combustion engine each piston has ignition at each second turn of the central shaft. This leads to the appearance of a very strong peak to 0.5X. At a 2-time internal combustion engine each piston has ignition in each turn of the central shaft. This leads to the appearance of a very strong peak in 1X. The vibration intense is very high in reciprocating engines. We always compare the new measurements with the previous ones in order to make an assumption. It is important for the measurements to be done under the same conditions.

The transient supply pumps have a smoother operation than compressors. If harmonics of the motion frequency of the piston exist that means that there is a leak to the piston.

We remind that:

Fans have b blades

Compressors have v vanes

Blade pass = BP = b · RPM

Vane pass = VP = v · RPM

Rpm = the speed of the shaft

The peak height increases when the gap between the vanes and the fixed hull is not constant. The peak height increases when there are problems to the flow, acute angles etc.

A turban is created by the air pressure or speed changes as it passes through the vanes of the fan or the compressor.

The turban creates random vibrations of low frequency, in the area of 50 CPM to 2000 CPM.

Cavitation creates random vibrations of high frequencies. It is shown as a curvature to the spectrum. Cavitation usually shows inefficient pressure in suction (low input pressure) or lack of fluid (starvation). Listening to the sound produced in the waveform, looks like there is sand in the pump.

The waveform is a very useful tool, as there are shown clearly, the energy explosions (blasts) of high frequency. Even that the electric engine 20 PS moves a centrifugal pump with 6 blades. If we notice the measurements to the waveform in horizontal direction, from the side of the pump, the energy explosions (blasts) are shown.

The curvatures and some coordination stimulations are seen to the spectrum.

If we see the measurements in vertical direction, the spectrum in G units with an emphasis at high frequencies, we see cavitation.

The analysis of gas or steam turbines is basically the same. The combustion chamber creates extend noise in gas turbines.

A displacement transducer is usually used.

(Non-contact, eddy current, proximity probe).

The spectrum that results with the displacement gives harmonics that are explained with the known way.

Turbines usually show a strong peak to the frequency BP = blade rate (number of blades to the shaft frequency). We check this in every measurement. BP frequency is high because the speed and the number of blades are also high. We must not forget the multilevel turbine engines.

If there is a fault in a blade or if their wear is not even, BP frequency will be modulated with the shaft frequency RPM and so there will be sidebands.

Turbines usually use journal bearings and so they may appear oil whirl and oil whip faults. We place two proximity probes in vertical directions (x and y) to depict the shaft's orbit.

The orbit of the signal in axis (x versus y) shows the motion of the shaft center inside the bearing. Imbalance, if it does not exist is shown with the perfect circle if the stiffness of the construction is the same in horizontal and vertical direction. The orbit is usually slightly elliptical. Misalignment causes increased ellipticity. The oil whirl creates the orbit wraps. Yet, there are also other reasons (friction to the bearing, looseness, coordination etc.), that cause faults.

10.9 FAULT DIAGNOSIS TO ROLLING BEARINGS

Shaft rubbing behaves similarly to looseness. Gives harmonics of 1X and often harmonics of 0.5X. Many times shaft rubbing becomes the stimulation of a coordination frequency. First of all we have harmonics. We notice harmonics of the electric current that we can overlook. Shaft rubbing developed stimulation in a coordination frequency, even in area 4X-5X and so the vibration intense here is higher than the vibration from other areas of the spectrum.

The waveform shows periodic explosions of transients while rubbing (we may also find sidebands due to the modulation phenomenon).

The time between two transients is equal to the period of rotation = time of the rotation of the shaft.

When issues occur to the grace of the journal bearings the spectrum will show what it has already shown for rotating looseness.

There will be high peaks to the harmonics of 1X. In most of the cases the vibration in the vertical direction will be greater than in the horizontal.

In extreme cases of severity of the phenomenon harmonics of frequencies of 0.5X or even X/3 will appear.

Oil whirl is a situation during which strong vibration among the frequencies 0.38X and 0.48X takes place. It never happens exactly in 0.5X.

It is caused by a great radial grace and slight loads and leads to oil collection which forces the shaft to move around the bearing.

That is a serious situation that must be fixed because leads to a metal to metal contact and rapid devastation of the construction.

Oil whip is a very devastating situation that can happen to big multistage systems, when they operate at speed higher than the crucial. It happens when frequencies of oil whirl coincide with a coordination frequency (natural frequency) of the shaft. When coordination occurs then high frequencies are released. Sometimes occurs at the start of the machines with large axis. The solution for both of the latest problems of the journal bearings is the small radial graces and the suitable radial loads. After the start of a large turbine engine it is significant to pass the crucial speed very quickly, before the Oil whip phenomenon manages to develop.

10.10 FAULT DIAGNOSIS TO ELECTRIC ENGINES

We will see the faults of AC (synchronous and inductive) and DC electric motors. Like other rotating machines so the electric motors show imbalance and misalignment problems.

We are going to see electric features of the engines and the faults they cause. There are 1phase and 3phase engines. 3phase engines are more sufficient and are used more. There are synchronous and inductive engines. Inductive ones are more common.

In an AC engine the stator bears the wraps into special corrugations. Three wraps form the two magnetic poles from the current action in a 3phase engine, which have a phase difference of 120º.

With a current line frequency of 50 Hz and a dipole of magnets (2 poles) the rotation of the magnetic field (i.e., the shaft) will have the same frequency as the line that means 50 Hz = 50 cycles per sec = 50×60 cycles per min = 3000 CPM = 3000 RPM. When we have 2 dipoles (i.e., 4 poles) we will have 6 wrappings and then the axis rotation will be done with half of the previous speed 1500 RPM.

When the rotor bears constant magnets, it is attracted to rotational movement by the attractive magnetic forces with the speed of the field. These speeds are similar (synchronous) and that is why the engine is called synchronous. An induction engine varies from a synchronous engine in that the rotor does not have constant magnets, but electric magnets i.e., conductive bars throughout its length (as cylinder hubs) that are evenly fixed in a socket and welded to the basis (rings) of the engine. This construction is similar to a hamster's or a squirrel's training cage. That is why many times these engines are called squirrel cage motors. Inductive currents that create tendency to the rotational construction of the rotor by reacting to the developing magnetic fields are created to the bars.

The inductive motor is many times called repulsive induction motor because it is based on magnetic repulsion. On the contrary, the synchronous electric motor relies on magnetic attraction.

If the friction to an inductive motor were zero, the rotor would rotate with the synchronous speed. There will always be a delay of the rotor to the synchronous speed due to the fact that friction and losses cannot be zero.

This delay is called sliding and depends on the load, as long as it is due to the developing inductive rows to the bars that increase with load. The vibration to electric motors is two times the frequency of the electric current (that is 50 Hz or 60 Hz) and has always a measuring peak height. It is created by the transient attraction between the rotor and the stator due to the presence of the transient magnetic field from magnetostriction.

Magnetostriction is the deformation of the magnetic material in the presence of magnetic field. It causes vibration in frequency two times the row frequency that is in frequency 100 Hz or 120 Hz to all electric devices (motors, generators, converters etc.).

In bipolar motors the synchronous speed 1X = 3000 CPM (or rotation frequency 1X = 50 Hz) the division between the harmonic 2X = 100 Hz and the magnetostriction which is two times the current frequency (50 Hz) is difficult, i.e., the harmonic is the 2X = 100 Hz, while the vibration by magnetostriction is made with 2×50 Hz = 100 Hz.

The same happens with the 60 Hz line.

In order to see if harmonic 2X actually occurs we set the machine to operate and take measurements first with load and then without load. The peak will disappear from magnetostriction but 2X remains.

In induction motors sliding gives the rotor a frequency lower than the line frequency (line frequency = LF = 50 Hz). Subsequently, rotor's 2X is always lower than 2 LF (Line Frequency) with which the rotor shows its problems (gap between stator and rotor).

Loosen or twisted bar foundation creates eccentricity to the rotor. The shaft eccentricity creates a transient air gap between the rotor and the stator. This gap is a pulsing vibration source.

$2 \times LF = 2 \cdot LF$ = twice the line frequency

PPF = pole pass frequency = sliding speed to the number of the poles.

Sliding speed is equal to the difference between the actual rotating frequency (of RPM) and the synchronous speed. We need measurements of high analysis to see these sidebands of PPF because the sliding speed is low, so PPF is also low.

Broken, cracked or worn rotor bars are a usual problem to induction motors, especially to these that start or stop with a load. The starting line is much larger than the operation line and creates inner tendency to the rotor bars that heats them.

Asymmetric heating bars of the rotor force it to deform or bend resulting in its imbalance (it has a peak to 1X). Local overheating may melt the bars. When we examine the rotor cooled the problem does not show up.

Broken bars may appear sidebands of PPF around 1X and the harmonics of 2X, 3X, 4X etc., depending on the seriousness of the problem. In an inductive motor with loosen bars to the rotor we will have a peak to the frequency RBF = rotor bar frequency with sidebands of twice the line frequency 100 Hz or 120 Hz.

RBF = number of the rotor's bars on the RPM.

LF = line frequency = 50 Hz or 60 Hz.

When a rotor slides on its shaft, high vibrations are caused and appear peaks in 1X and its harmonics. The phenomenon is actuated by the sharp load or line voltage changes. When the wrappings in the rotor motor are even a little loosen, the vibration in a frequency twice the line frequency will increase. This problem is destructive because through friction the cable insulation is removed with the result of short-circuits and faults to the stator. The rotor as well as the stator are made by blades (thin foils) which are insulated from each other. If the blades are short-circuited local overheats and material warps develop. Short-circuited blades cause vibration in a frequency equal to twice the line frequency. Warps cause appearance of peak in 1X with sidebands of PPF.

Issues related to the phases due to loosen couplings may cause a high vibration to the double line frequency (2×LF) with sidebands of a third of LF.

If the line frequency is equal to 50 Hz and we find the sideband of the twice sliding frequency 2×Slip to the spectrum, then the real speed of the engine results.

Example

Even that the line frequency LF is equal to 60 Hz.

If the sliding speed is 40 CPM i.e., Slip = 40 CPM = 0.666 Hz or 2×Slip = 1.333 Hz

Then the peaks of the sidebands will appear in frequencies 58.666 Hz and 61.334 Hz.

10.11 SYMPTOMS OF FAULTS

1. Ski-slope

Symptom: **High peak in frequency close to 0 Hz, with a decreasing intension as the frequency increases.**

If we notice that the spectrum starts with a high peak from the slight frequencies and gradually the peak height is decreased as we proceed to higher frequencies, then the sensor may have either a functional problem or may have been hit during the measuring procedure, or a transient phenomenon occurred on it. The transient phenomenon may be of mechanical origin (shock or vibration) or of thermal origin (the sensor contacted with a too cold or too hot surface) or of electric origin. There will be the ski-slope curve and noise at very high frequencies when the sensor is saturated, which happens when there is a high vibration source at high frequencies.

2. Increased general noise

Symptom: **Lifted spectrum base**

If there is increased general noise, we probably have great wear to a bearing. If this noise is detected to the side of spectrum high frequencies, then cavitation may occur. Small curvatures (lifts) of the spectrum base are rather due to coordination or sidebands located very close to each other. In that case we should make a high analyzed measurement or zoom and use a logarithmic scale to clarify which cause occurs. If we are able to change the machine velocity the coordination frequency won't change, as the other peaks with their side frequencies will be removed.

3. Poor balancing, lack of balancing, imbalance.

Symptom: **Radial (V & H) peak in 1X**

Lack of balancing indicates the situation during which the rotation axis of the shaft and the axis center of gravity do not coincide. In other words, the center of gravity is not on the rotation axis of the shaft, therefore an eccentric mass is detected along the shaft.

4. Static imbalance

Symptom: **Radial (V and H) peak in 1X**

In a shaft with imbalance, we expect to see the waveform in sinusoidal curve, with frequency equal to the rotation frequency of the shaft and a high peak to the spectrum in position 1X of the shaft. The simplest imbalance formation is represented by a mass detected in a point of the shaft. This case is called static imbalance because it is found even when the shaft does not rotate (when the shaft is placed in a horizontal position to a zero-friction bearing, the detected mass will turn the shaft, so that the mass due to weight will occupy the lowest possible position). Static imbalance leads to peaks in position 1X on both bearings of the shaft, the forces on both bearings are always towards the same direction, that is the signals received from the shock measurement on both bearings are in phase.

5. Couple imbalance

Symptom: **Radial (V & H) peak in 1X**

A shaft with couple imbalance does not have static imbalance, i.e., always balances during static imbalance examination. Yet, when a couple imbalanced shaft rotates, it develops centrifugal forces on the bearings which are in opposite directions, i.e., the signals received from the shock measurement on both bearings have a phase difference of 180^{o}.

6. Imbalance of components in cantilever

Symptom: **High axial peak in 1X**

Radial (V & H) peak in 1X

When a shaft has a component in cantilever (motor, pump, fan, small turbine engine, etc.), it develops a peak in position 1X to all directions: axial, horizontal, vertical. This happens because the cantilevered position of the component creates, during operation, a bending arrow, which causes imbalance. This in turn develops bending moment to the shaft, which the bearing takes. The developing force of the couple acts axially and tends to move the hull of the bearing axially.

7. Imbalance in vertical shafts

Symptom: **Radial (H) peak in 1X**

A vertical imbalanced shaft will indicate peak in position 1X when the measurement is done radially at any horizontal position. In a vertical motor-pump set we must loosen the middle elastic coupling in order to find the imbalance (if it is in the motor or the pump). Then we can run the engine alone and measure the shock in position 1X. If the peak remains high, then the engine has the problem. If there is no peak, then the pump has the problem.

8. Component with eccentricity

Symptom: **radial (V and H) peak in 1X**

When the barycenter is off rotation axis there is eccentricity as happens to cogwheels, bearings and shafts.

The components with eccentricity develop high peaks to radial vibration components in position 1X and copy static imbalance.

9. Pulleys with eccentricity

Symptom: **Radial (V and H) peak in 1X**

When the geometrical center of the pulley is off its rotation center then the pulley works with eccentricity. Eccentric pulleys create high peaks in position 1X in the radial direction and especially in a parallel direction with the belts. This situation copies static imbalance. The phenomenon is noticed to the shaft of the drive and also the driven pulley. That is, a high peak will appear to the bearing of the drive pulley in position 1X, as also it will appear in the position of its frequency. The check about which pulley is problematic is done by removing the belt and measuring the shocks of the engine's bearing. If the peak in 1X stays the problem has the driving pulley. If the peak disappears, the problem is at the driving pulley.

10.Misalignment. Lack of alignment

Symptom: **Axial peak in 1X**

Radial (V and H) peak in 2X

Parallel and angular misalignment.

In most of the cases both types exist.

General rule: diagnosis is based on the shock measurements in the radial **(V and H)** directions in position 2X but also to the high peak intense in the axial direction in position 1X.

11.Angular misalignment

Symptom: **Axial peak in 1X**

Low axial peak in 2X

Low radial (V and H) peak in 1X

Angular misalignment creates bending tendency to each shaft. This tendency develops on both bearings a strong axial vibration in frequency 1X and a smaller axial vibration in frequency 2X.

Enough vibration is also developed in the radial direction **(V and H)** in frequency 1X (and sometimes a smaller in frequency 2X). On both sides of the shafts' couplings, the axial measurements have a phase difference of 180º while the radial measurements are in phase.

12.Parallel misalignment

Symptoms: **Radial peak (V and H) in 2X**

Small radial (V and H) peaks in 1X

Parallel misalignment creates a shear and bending tendency at the edges of united coupled shafts. So, we have strong peaks in frequency 2X in the radial directions **(V and H)** at the closer to bearing coupling of each shaft. We usually also have lower peaks in frequency 1X in the radial directions **(V and H)** of the bearings above.

Axial measurements will give low peak values in positions 1X and 2X.

Both radial and axial vibrations have a phase difference of 180º to bearings on both sides of the coupling.

13.Shaft with a bending arrow

Symptoms: **Axial peak in 1X**

A shaft with a bending arrow mainly produces shocks in radial direction in frequency 1X when the bending arrow occurs in the middle of the shaft. Yet, if the bending occurs near the coupling, we will see a peak in position 2X in the radial direction. There are low peaks in the radial directions (H and V) on both positions 1X and 2X. The shock measurements in the axial direction in position 1X (on both sides of the arrow) have a phase difference of 180º.

14. Triangular (chocked) bearing

Symptom: **axial peak in 1X**

Axial peak in 2X

Axial peak in 3X

Such a problematic bearing is a misaligned bearing and will create significant axial vibration in positions 1X, 2X and 3X. May be considered that symptoms are due to misalignment or imbalance of cantilevered component because of high axial vibration production. But the existence of peaks in frequencies 2X and 3X indicates that there are no other causes.

15. Looseness due to rotation

Symptoms: **harmonics of radial peak in frequency 1X**

Harmonics of radial 0.5X in a large shock

Rotation looseness is caused by the excessive grace between the rotating and the constant part, as in the bearings. Between two constant components, such as the machine foundation or the supporting table of a bearing and the machine hull, the looseness develops by other causes except for rotation. Great radial grace at sliding bearings and rolling bearings leads to the development of harmonics in 1X that most of them reach up to 10X. Excessive radial grace at sliding bearings produces harmonics of 0.5X called half order harmonics. They can be caused by shaft rubbing on the bearing and strong collisions and in some cases harmonics of 1/3X may also be created.

16. Construction looseness

Symptoms: **Harmonics of radial peak in frequency 1X**

Construction looseness is looseness between a machine and its bearing that increases the vibration towards the direction of the low solidity, which is usually in the horizontal radial position, on horizontal machines, towards which horizontal bearings have a great elasticity. Small order harmonics of 1X are present in cases of strong construction looseness. The discrimination between looseness or elasticity of foundation and imbalance is difficult in vertical machines. When the horizontal peak in 1X is higher than the vertical peak in 1X then rather looseness is to blame. When the horizontal peak in 1X is lower or equal to the vertical peak in 1X then rather imbalance is to blame. Looseness or elasticity of the foundation is caused by loosen bolts, wear and crack of the cooperating surfaces. A machine with elastic couplings has always a higher vibration in the horizontal direction. A vibration measurement in the vertical direction will give a phase difference of 180º between the machine and the foundation.

17. Coordination

Symptom: **lifted spectrum only in one direction**

Coordination is a functional condition where the stimulating frequency is near the natural frequency of the construction, with which the construction performs oscillation. There is usually a high peak on the lifted area of the spectrum only in one direction. If we alter the operation frequency so that it will be removed from the construction's natural frequency, then the peak decreases significantly the coordination ceases to exist, and the construction is not stimulated.

A very primary ingredient of fault diagnosis is the discovery of coordination. Coordination is the situation in which the frequency of a vibration source is close to the natural frequency of the machine or a component, so oscillation will occur with the component's or the machine's natural frequency. Many times, natural frequencies are also called crucial frequencies or crucial velocities. A construction can have many natural frequencies. When coordination occurs, the vibration that results can have a high intense and cause rapid destruction. Coordination is shown to the spectrum as a peak in constant frequency even when the machine velocity alters. The peak can be very sharp wide, as this depends on the efficiency of the construction's damping to the given velocity. If the coordination frequency coincides with 1X or 2X or the blade frequency, then the vibration will increase and will be shown as a high peak at the top of a system of lower peaks curvature.

We hit the machine with a stick or rubber and take the shock measurements. The vibration will be done with the machine's natural frequency. During the machine start and by increasing the velocity or during the decrease of velocity from the maximum up to zero, the collection of measurements will indicate the peaks to the spectrum in the positions of the machine's coordination frequencies. If the machine changes speed, we can monitor the waveform. The oscillation amplitude will become maximum when the machine turns with its coordination speed (we can measure with a tachometer). There is a phase difference of 90º between the vibration source that will be stimulated and the construction's response with the natural frequency.

18. Bearings

Symptom: **peak with harmonics in non-synchronous frequencies**

Faults related to the bearings follow conventional indications and symptoms with the ringing of the bearing in high frequency as a principle. Various tools, such as Spike energy etc. detect the fault's start on the bearing even in a very early stage. Even Demodulation method can also be used. As the fault to the bearing gets bigger the spectrum will be change in a characteristic way. Peaks will appear in non-synchronous frequencies (e.g., 3.9X or 6.45X etc.), that will be followed by their harmonics and sidebands with a difference of 1X will appear.

19.Pumps, fans, compressors

Symptom: **Peak in vane pass or blade pass frequency**

Pumps, fans, and turbine engines show peak in “blade-pass” or “vane-pass” frequency, which is equal to the product of the number of the blades to the shaft’s frequency. The size of the peak increases due to the gap change between the blades and the constant hull and due to flow obstacles, or due to sharp direction changes of the flow.

20.Turbulent flow

Symptom: **Random vibration in the area of 1Hz up to 35 Hz**

As the flow passes through the pump or fan blades etc., bears changes to the pressure or speed, which create vibrations in the area of **1Hz up to 35 Hz** when they cause turbulent flow.

21.Cavitation

Symptom**: Noise in high frequencies**

Cavitation indicates low suction pressure (low input pressure to the pump) or lack of fluid. Cavitation causes random vibration with the form of noise in the high frequencies of the spectrum. It appears in the form of lifted spectrum base. If we listen, the waveform sounds like there is a grid in the pump.

22.Reciprocating engines

Symptom: **2- stroke engine = peak in frequency 1X**

4- Stroke engine = peak in frequency 0.5X

Shock in reciprocating engines is very large.

A 4-stroke engine has ignition in each second shaft rotation. The period of the shaft is T = 1/X. this means that the ignition (so the impact) happens once per two periods, i.e., with frequency equal to 1/2T = 0.5/T = 0.5X. A 2-stroke machine has ignition in each rotation of the shaft. The shaft period is T = 1/X. this means that ignition (and so the impact) happens once per period, i.e., with a frequency equal to 1X.

23.Eccentricity to the stator of the motor

Symptom: **Radial peak at twice the frequency of the electric line i.e., in 100 Hz**

The stator's issues create high vibration in a frequency that is twice the line frequency i.e., in 100 Hz. The stator's eccentricity creates an asymmetric air gap between the vanes and the stator that produces shocks to a given direction. Soft or crumbled engine legs or twisted foundation have as a result the stator's eccentricity.

24. Eccentricity to the motor's rotor

Symptoms: **Radial peak in frequency 100 Hz**

Pole pass frequency around 1X

An eccentric rotor creates transient air gap between the blades and the stator. This results to a pulsing shock source. Beyond the peak to the twice frequency of the line 100 Hz, we will expect to see sidebands of pole pass frequency around the peak in frequency 1X. This frequency is equal to the sliding frequency to the number of poles. The sliding frequency represents the difference between the real engine speed and its synchronous speed.

25. Crumbled rotor

Symptom: **Radial peak in frequency 1X**

Asymmetric heat of the rotor due to unequal line distribution to the rotor's bars causes twist of the rotor, with the result of imbalance creation. The symptoms do not exist when the motor is cool. Local overheating to motors can cause violent conditions to the bars, so that a bar overheated too much so it melts resulting to the development of an air gap (as in the previous cases).

26. Cracked or broken bar of the motor

Symptoms: **pole pass sidebands around frequency 1X**

Appearance of harmonics.

Cracked motor bar creates pole pass frequency sidebands around the frequency 1X and the harmonics of 2X, 3X etc., the pole pass frequency is equal to the sliding frequency to the number of poles. The sliding frequency represents the difference between the real speed of the engine and its synchronous speed. An inductive engine with a fault to the bars gives a characteristic oscillation of variable amplitude with a frequency twice the engine's sliding frequency. This phenomenon is called beating, which we can measure and listen to. The frequency and the oscillation amplitude beating depend on the sliding frequency that relies on the engine load.

27.Loosen bars of the engine

Symptom: **100 Hz sidebands around the frequency RBF**

If there are loosen bars to the engine a peak will appear in frequency RBF = Rotor Bar pass frequency (which is equal to the product of the bars to the rotor's frequency) and around it will appear sidebands of twice the line frequency (100 Hz). Even when we do not know the number of bars of the engine, if we notice peak in high frequency and various sidebands around it with differences of 100 Hz, we can accord this frequency to the specific fault.

28.Loosen rotor

Symptom: **High peak in 1X and existence of its harmonics**

Sometimes the rotor can slide on the shaft, with breaks depending on the temperature. This causes strong vibration in frequency 1X and its harmonics. This condition is incited by sharp load or line voltage changes.

29.Loosen wrappings to the rotor

Symptom: **High radial peak in frequency of 100 Hz**

If the wrappings loosen even a little, the peak in the twice (100 Hz) line frequency (50 Hz) will increase. This situation is destructive because it destroys (rubs) the conductors' insulation and leads to defective wrappings that end to a fault of the stator.

30.Problematic blades

Symptom: **high radial peak in frequency 100 Hz**

The rotor and the stator of alternative current motors (AC) are made by thin blades that are insulated to each other. If the blades short-circuit (stop being insulated) a local overheat and twist will occur. Such blades create high peak in the twice (100Hz) line frequency (50 Hz) while the twist increases the peak in frequency 1X and often appears pole pass sidebands in it.

31.Cogwheels

Symptom: **peaks in shafts frequencies and to Gearmesh**

In cooperating cogwheels, we will find peaks (maybe low) in the shafts' frequencies and in the gearmash frequency of the teeth. Perhaps a peak in frequency 2X exists and sidebands of a shaft around the gearmash frequency appear. Gear mesh = Number of teeth × shaft speed. Peaks in these frequencies will be in the radial direction on wheels with straight teeth and in axial direction on wheels with helical teeth. The waveform analysis is a very strong tool when it comes to examine measurements from cogwheels boxes. Each tooth, as it cooperates, produces a pulse to the waveform. We can study the waveform and measure the number of teeth as the time (speed) difference of the one tooth from its near is equal to 1X divided by the number of teeth. Depending on the fault's nature, we can see to the waveform a pulse per rotation different from the others (e.g., it gives minimum amplitude if it is missing).

32. Wear on tooth

Symptom: **1X sidebands around gear mesh frequency**

When the wear on teeth begins the following happens:

At the beginning the peaks to the sidebands increase around the frequency of the cogwheel shaft that has the wear problem. The second thing that happens is that the natural frequency of the cogwheel begins to get stimulated. This peak will have a wider base to the spectrum and will show sidebands.

33. Tooth load

Symptom: **high peak in gear mesh frequency**

The size of peak in gear mesh frequency depends on the wheel shafts alignment and the size of the gear load. Therefore, the existence of a high peak in the gear mesh peak does not definitely mean the existence of a problem on cogwheels.

34. Gear profile grace (backlash)

Symptom: **1X sidebands around gear mesh frequency**

Backlash phenomenon to cogwheels will cause 1X sidebands (of the shaft's frequency) around gear mesh frequency. When a backlash occurs and the load increases, then the peak in gear mesh and the peak in the natural frequency of the machine will decrease.

35. Cogwheels with eccentricity

Symptom: **1X sidebands around gear mesh frequency**

Cogwheels with eccentricity and wheels in a shaft with a bending row cause sidebands of the problematic wheel's shaft around gear mesh frequency. Many times, we only see one sideband.

36. Misaligned cogwheels

Symptoms: **1X sidebands around gear mesh frequency**

Appearance of gear mesh frequency harmonics

Misaligned cogwheels cause high peaks in the gear mesh frequency and create sidebands. They often cause harmonics of the gear mesh frequency with high peaks in the double and triple harmonic of gear mesh. Consequently, is recommended to set the frequency range (F_{max}) quite large to include these frequencies to the spectrum.

37. Cracked or broken tooth

<u>Symptom:</u> **high radial peak in frequency 1X**

Peak in the natural frequency of the wheel

1X sidebands around gear mesh frequency

A cracked or broken cogwheel tooth causes the following:

High peak in rotation frequency of the wheel. It stimulates the natural frequency of the wheel. It develops sidebands around frequency gear mesh. Yet, the best way to see if the effect of such a tooth is through the waveform. If the wheel has 13 teeth, one of the 13 pulses of the waveform will be different. We have 12 small pulses in the waveform and a great one. The time difference between the great pulses is equal to the period of the wheel's shaft because the broken tooth contacts once per each rotation.

38. Hunting tooth frequency

<u>Symptom:</u> **hunting tooth frequency**

Hunting tooth frequency is the frequency with which a specific tooth of one wheel will cooperate with a specific tooth of the other wheel. If the transmission relation is an integer number 1, 2, 3 etc., the hunting tooth frequency is equal to the frequency of the big wheel, because we will have hunting of the teeth in each rotation of the big wheel. This causes uneven teeth wear (wear in specific teeth) because there is a repeating contact of the one wheel's same teeth with the other wheel's same teeth, which causes this detected wear. In a couple of cooperating teeth, the hunting frequency is found when we divide the gear mesh frequency with the minimum common multiple of the teeth (when the numbers of teeth are first to others then the minimum common multiple is their product).

39. Couplings

<u>Symptom:</u> **peaks in frequencies 1X and 2X**

When the coupling is not succeeded (e.g., the flanges' faces are not parallel) a vibration similar with this of the angular misalignment is caused. There is usually a poor balancing of the coupling that causes peaks in frequencies 1X and 2X in the radial directions. Worn couplings have all the symptoms of misalignment and loosen construction.

40. Worn or loosen belts

<u>Symptom:</u> **belt rate with sidebands**

When the belt is worn or loosen peak will appear in frequency BR belt rate and harmonics with the double frequency of belt rate. The basic frequency is called belt rate or fundamental belt pass frequency and indicates frequency with which a point of the belt passes by the specific constant point of reference. BR is always lower than each pulley's frequency. It is calculated as follows:

1 = drive pulley, 2 = driving pulley

n = speed, d = diameter, $d_2 = d_1 n_1 / n_2$

BR = $\pi \cdot d_1 n_1 / Belt\ length$ where $\pi = 3.1416$

41. Pulleys with eccentricity

<u>Symptom:</u> **high radial peak in frequency 1X**

High axial peak in frequency 1X

A pulley with eccentricity will give a strong peak in frequency 1X in the axial direction that is parallel to the fields of the belt or to the component of the belt's forces. In the belt-pulley system the eccentricity of the pulley that is shown in the peak 1X, will also be shown in the 1X of the other pulley. Pulleys' misalignment causes high vibration in 1X mainly in the axial direction. Sometimes this may also develop harmonics of BR of the belt in the axial direction.

42. Belt's coordination

<u>Symptom:</u> **high radial peak in frequency 1X**

When the coordination frequency of the belt (natural frequency) coincides with one of the two pulleys' frequencies (drive or driving) then we have coordination with the belt. It may cause a high peak. The belt's coordination frequency changes either with the change of its tensile strength (through changing the distance of the centers), or with the placement of a new belt.

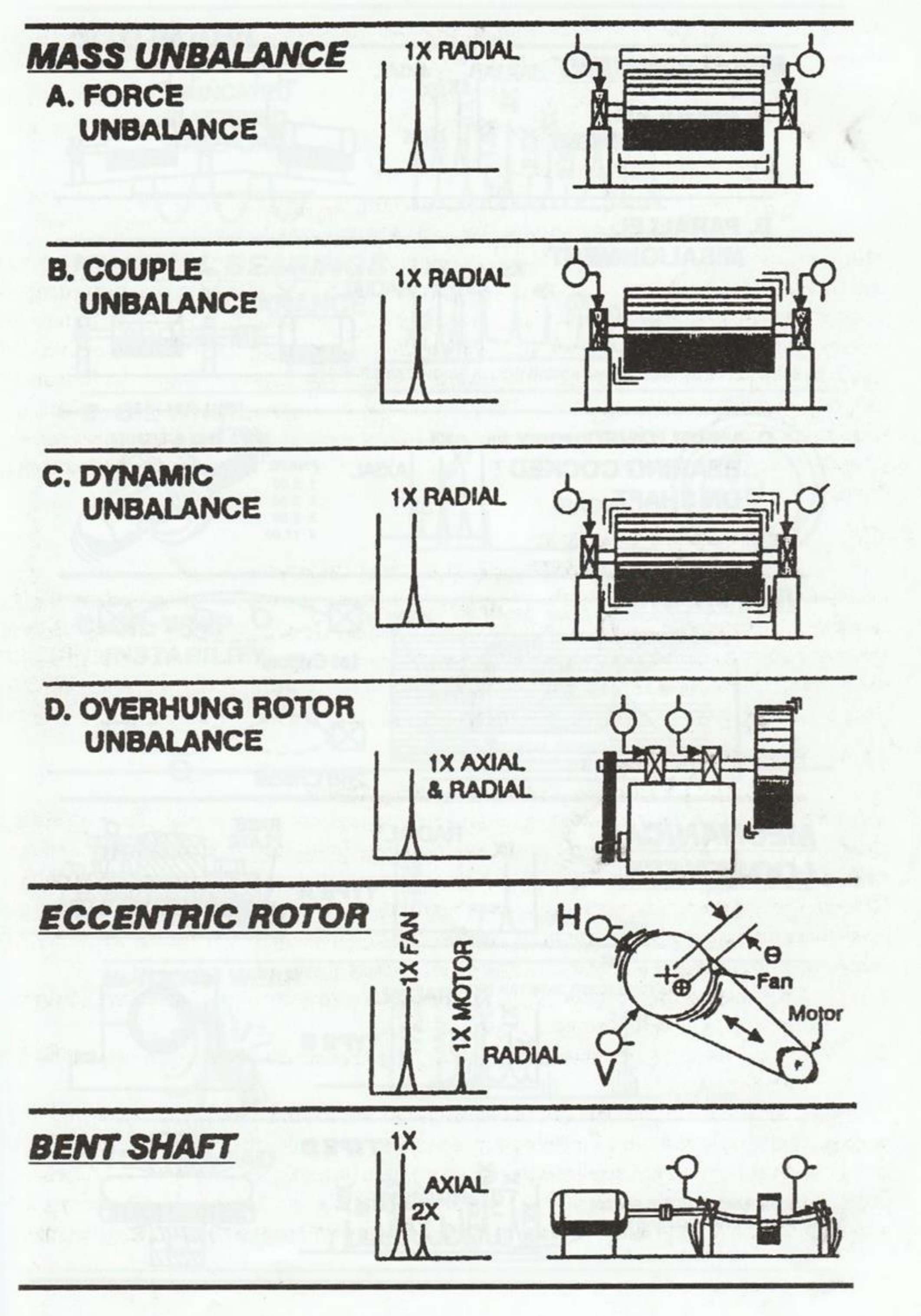
43. External noise

<u>Symptom:</u> **Peak in a non-synchronous frequency**

We may detect to the spectrum a peak in frequency with an unknown origin. Before we conclude, we must examine the case this specific shock peak origins from a near machine. The peak will belong to a non-synchronous frequency, and it would be possible if it was identified with some of the frequencies of the machine. There are two ways to proove if the specific vibration comes from another machine:

1. We shut the machine or change its speed and consider if the strange vibration is still there.
2. We examine the neighbor machines, if the strange frequency belongs to someone.

If we find out that this frequency belongs to a near machine, perhaps we must not ignore it. For example, if the machine in which we detected the strange frequency is stationary for a long time (maybe is a stand-by machine), then the transferring vibration from the near machine can cause the brinelling procedure to the bearings of the stationery machine.


MASS UNBALANCE
A. FORCE UNBALANCE
1X RADIAL
B. COUPLE UNBALANCE
1X RADIAL
C. DYNAMIC UNBALANCE
1X RADIAL
D. OVERHUNG ROTOR UNBALANCE
1X AXIAL & RADIAL
ECCENTRIC ROTOR
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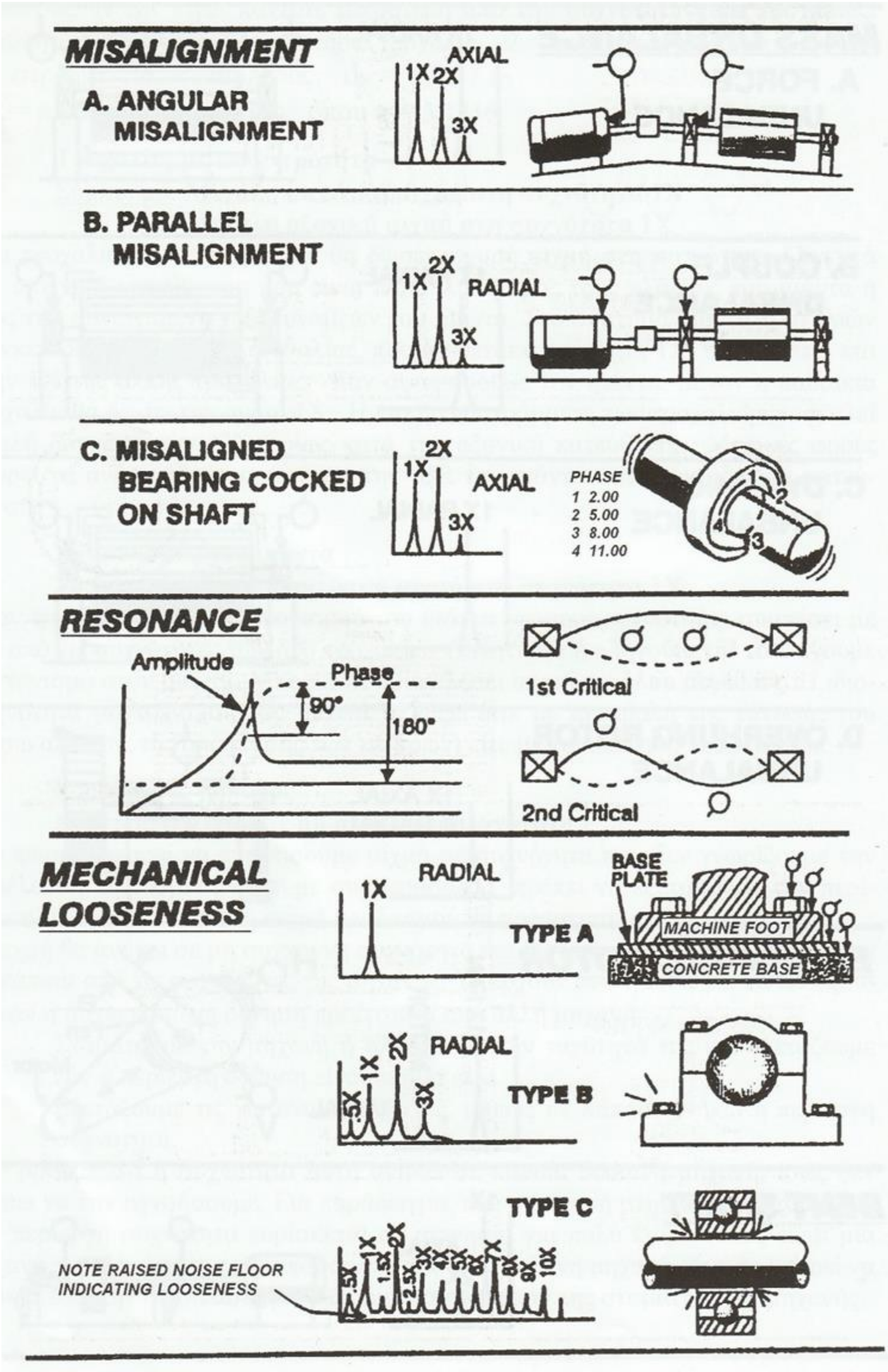

MISALIGNMENT
A. ANGULAR MISALIGNMENT
AXIAL
1X
2X
3X
B. PARALLEL MISALIGNMENT
RADIAL
1X
2X
3X
C. MISALIGNED BEARING COCKED ON SHAFT
AXIAL
1X
2X
3X
PHASE
1 2.00
2 5.00
3 8.00
4 11.00
RESONANCE
Amplitude
Phase
90°
180°
1st Critical
2nd Critical
MECHANICAL LOOSENESS
RADIAL
1X
TYPE A
BASE PLATE
MACHINE FOOT
CONCRETE BASE
RADIAL
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TYPE B
TYPE C
NOTE RAISED NOISE FLOOR INDICATING LOOSENESS
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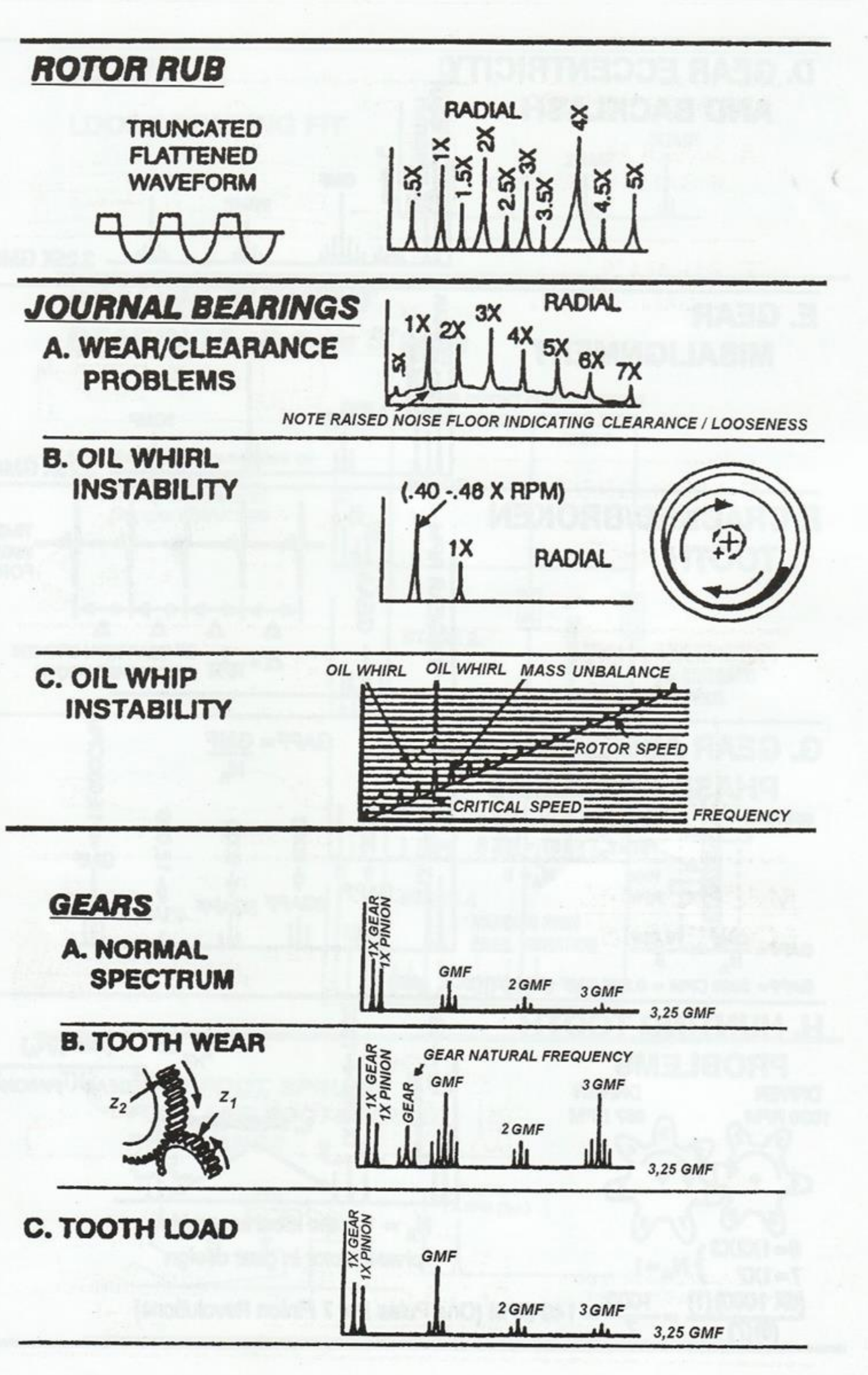
ROTOR RUB
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4X
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JOURNAL BEARINGS
A. WEAR/CLEARANCE
PROBLEMS
RADIAL
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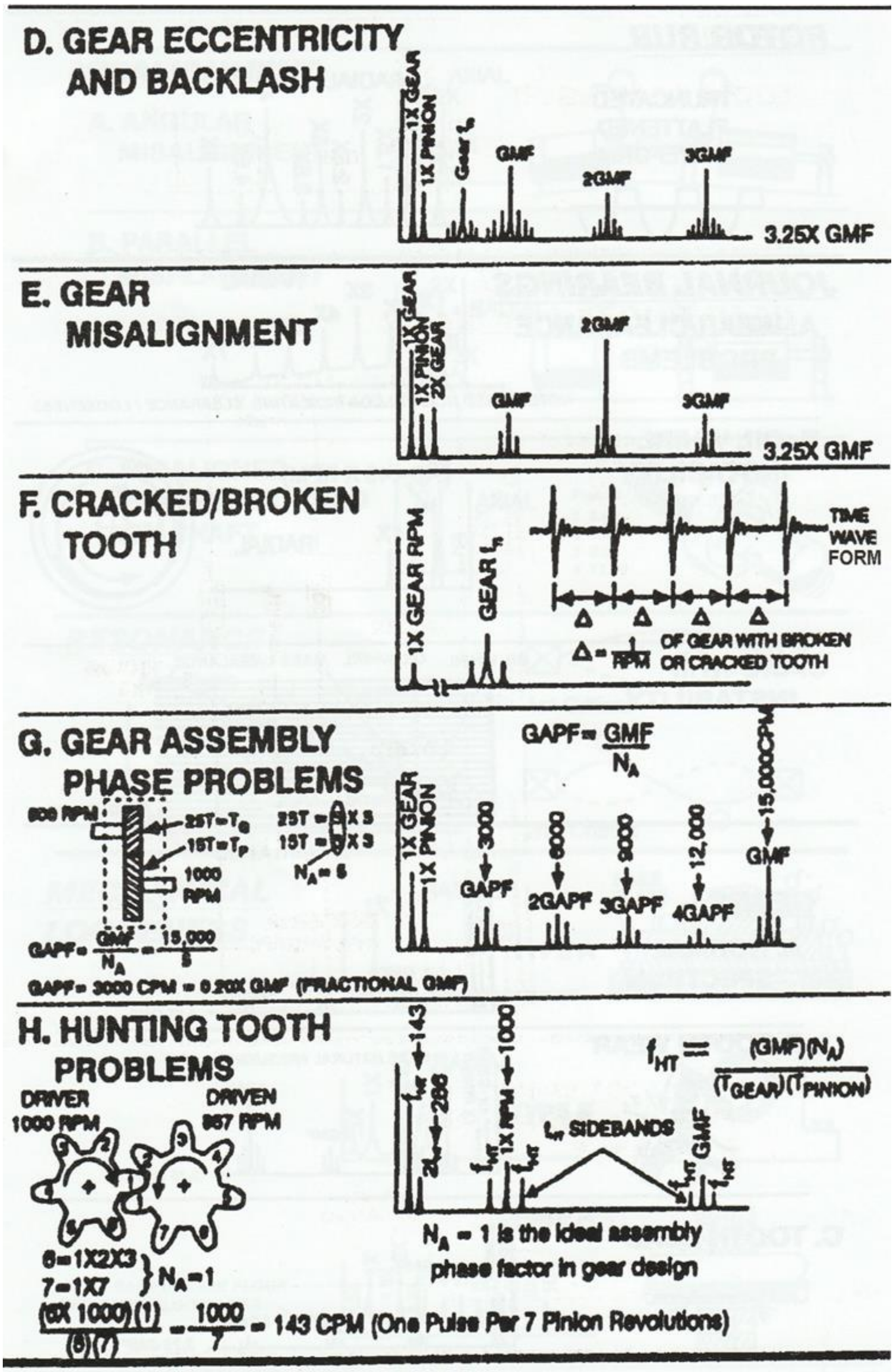
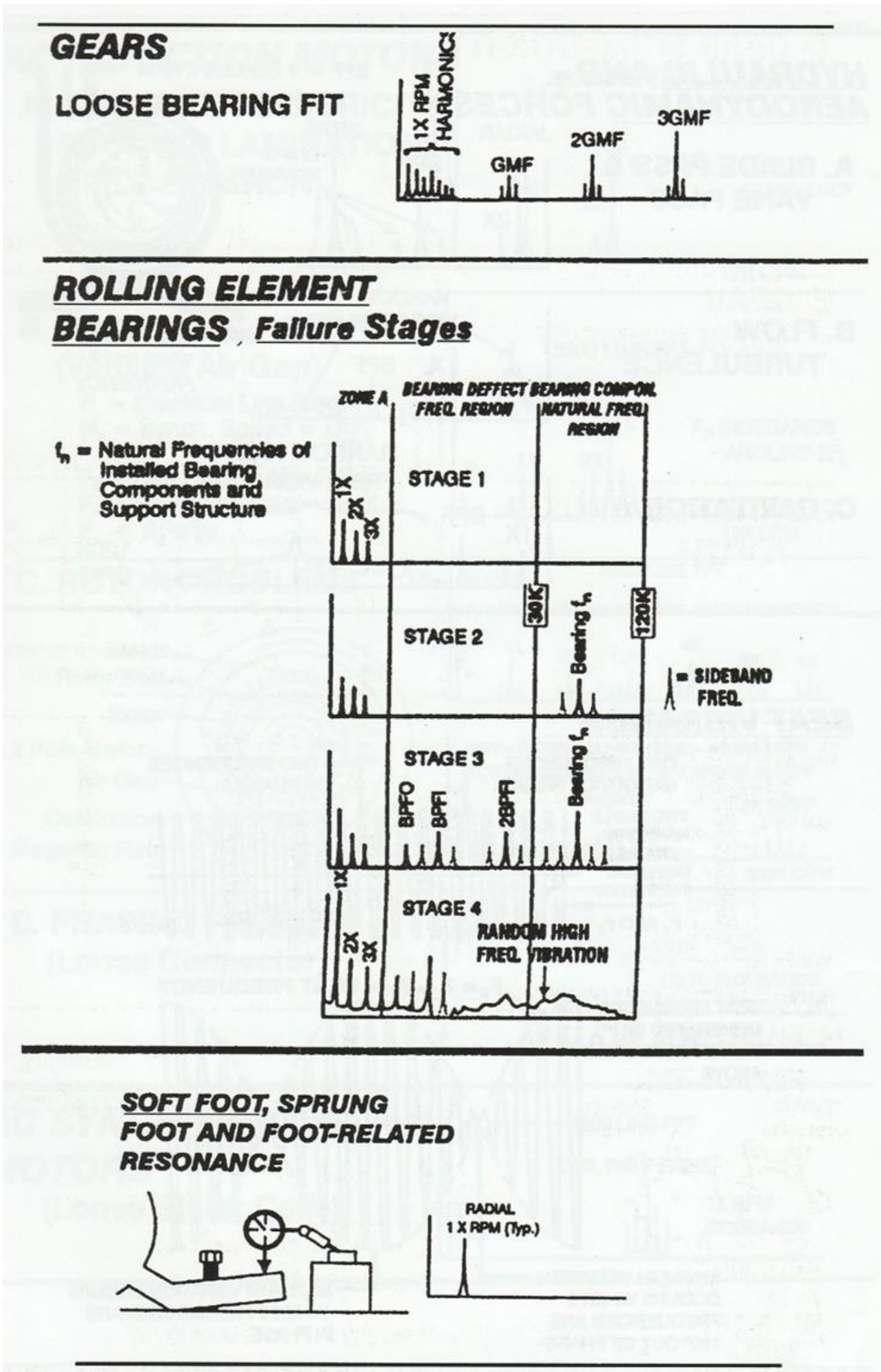
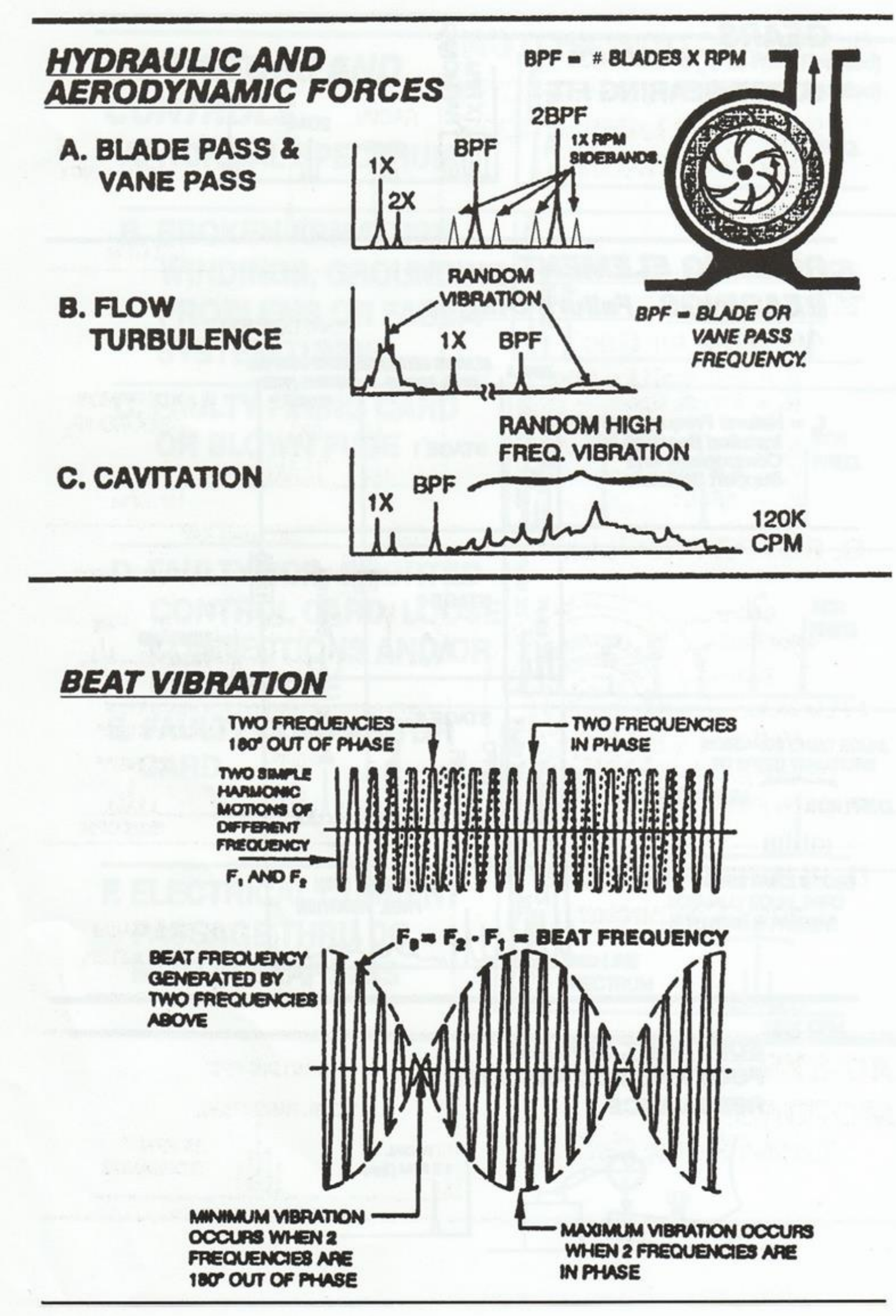
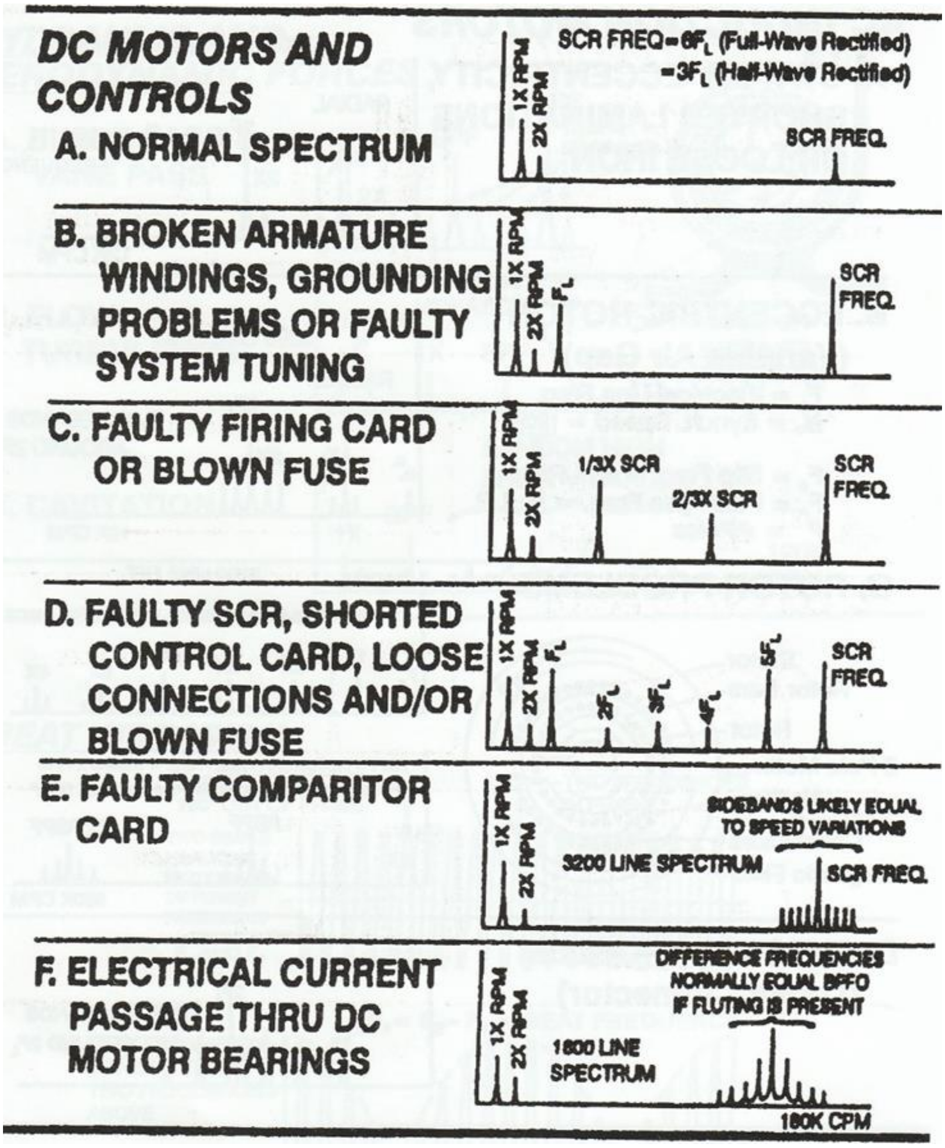
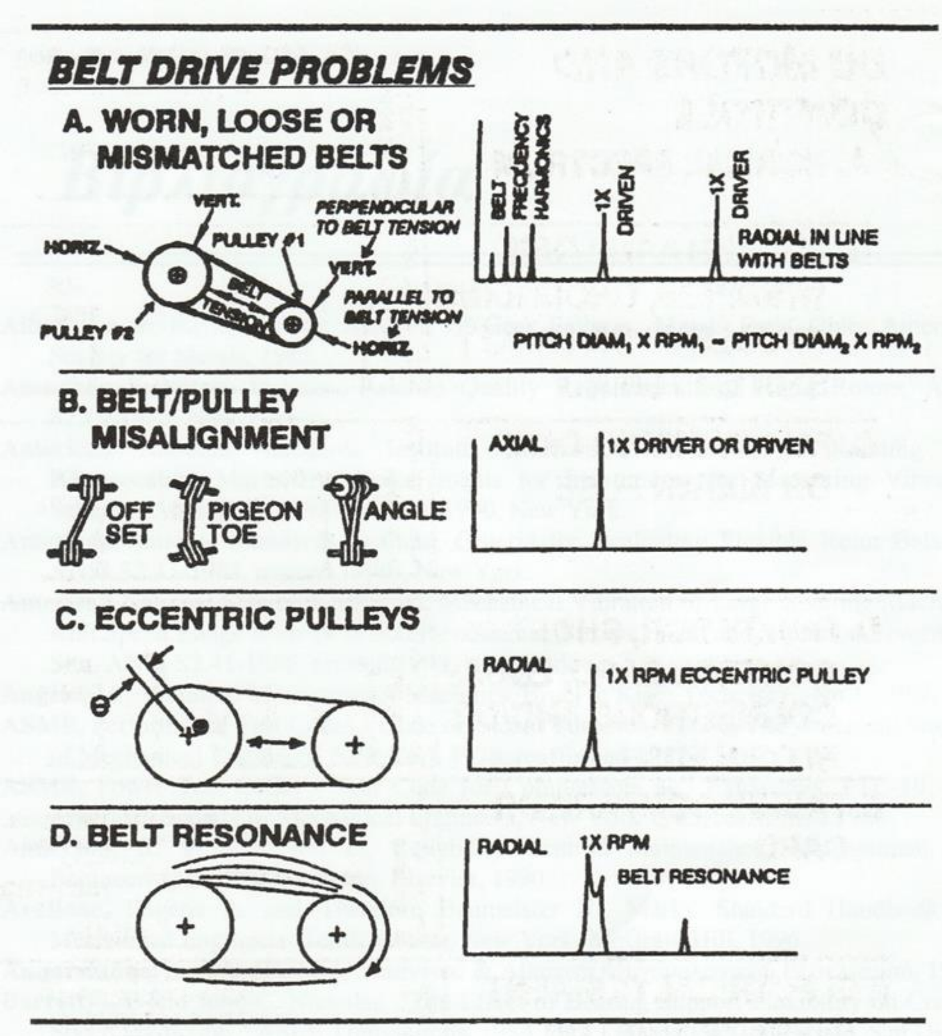
6X
7X
NOTE RAISED NOISE FLOOR INDICATING CLEARANCE / LOOSENESS
B. OIL WHIRL
INSTABILITY
(.40 -.48 X RPM)
1X
RADIAL
C. OIL WHIP
INSTABILITY
OIL WHIRL
OIL WHIRL
MASS UNBALANCE
ROTOR SPEED
CRITICAL SPEED
FREQUENCY
GEARS
A. NORMAL
SPECTRUM
1X GEAR
1X PINION
GMF
2 GMF
3 GMF
3,25 GMF
B. TOOTH WEAR
GEAR NATURAL FREQUENCY
1X GEAR
1X PINION
GEAR
GMF
2 GMF
3 GMF
3,25 GMF
C. TOOTH LOAD
1X GEAR
1X PINION
GMF
2 GMF
3 GMF
3,25 GMF


D. GEAR ECCENTRICITY AND BACKLASH
1X GEAR
1X PINION
GMF
2GMF
3GMF
3.25X GMF
E. GEAR MISALIGNMENT
1X GEAR
1X PINION
2X GEAR
GMF
2GMF
3GMF
3.25X GMF
F. CRACKED/BROKEN TOOTH
1X GEAR RPM
TIME WAVE FORM
OF GEAR WITH BROKEN OR CRACKED TOOTH
G. GEAR ASSEMBLY PHASE PROBLEMS
1000 RPM
GAPF = 3000 CPM = 0.20X GMF (FRACTIONAL GMF)
1X GEAR
1X PINION
GAPF
2GAPF
3GAPF
4GAPF
GMF
15,000CPM
H. HUNTING TOOTH PROBLEMS
DRIVER
1000 RPM
DRIVEN
857 RPM
1X RPM
1000
GMF
SIDEBANDS
N = 1 is the ideal assembly phase factor in gear design
143 CPM (One Pulse Per 7 Pinion Revolutions)


GEARS
LOOSE BEARING FIT
1X RPM HARMONICS
GMF
2GMF
3GMF
ROLLING ELEMENT BEARINGS, Failure Stages
ZONE A
BEARING DEFFECT FREQ. REGION
BEARING COMPON. NATURAL FREQ. REGION
f_n = Natural Frequencies of Installed Bearing Components and Support Structure
STAGE 1
1X
2X
3X
30K
120K
STAGE 2
Bearing f_n
= SIDEBAND FREQ.
STAGE 3
BPFO
BPFI
2BPFI
Bearing f_n
STAGE 4
1X
2X
3X
RANDOM HIGH FREQ. VIBRATION
SOFT FOOT, SPRUNG FOOT AND FOOT-RELATED RESONANCE
RADIAL
1 X RPM (Typ.)


HYDRAULIC AND AERODYNAMIC FORCES
A. BLADE PASS & VANE PASS
1X
2X
BPF
2BPF
1X RPM SIDEBANDS.
BPF = # BLADES X RPM
BPF = BLADE OR VANE PASS FREQUENCY.
B. FLOW TURBULENCE
RANDOM VIBRATION
1X
BPF
C. CAVITATION
RANDOM HIGH FREQ. VIBRATION
1X
BPF
120K CPM
BEAT VIBRATION
TWO FREQUENCIES 180° OUT OF PHASE
TWO FREQUENCIES IN PHASE
TWO SIMPLE HARMONIC MOTIONS OF DIFFERENT FREQUENCY
F1 AND F2
FB = F2 - F1 = BEAT FREQUENCY
BEAT FREQUENCY GENERATED BY TWO FREQUENCIES ABOVE
MINIMUM VIBRATION OCCURS WHEN 2 FREQUENCIES ARE 180° OUT OF PHASE
MAXIMUM VIBRATION OCCURS WHEN 2 FREQUENCIES ARE IN PHASE


DC MOTORS AND CONTROLS
A. NORMAL SPECTRUM
1X RPM
2X RPM
SCR FREQ = 6F_L (Full-Wave Rectified)
= 3F_L (Half-Wave Rectified)
SCR FREQ.
B. BROKEN ARMATURE WINDINGS, GROUNDING PROBLEMS OR FAULTY SYSTEM TUNING
1X RPM
2X RPM
F_L
SCR FREQ.
C. FAULTY FIRING CARD OR BLOWN FUSE
1X RPM
2X RPM
1/3X SCR
2/3X SCR
SCR FREQ.
D. FAULTY SCR, SHORTED CONTROL CARD, LOOSE CONNECTIONS AND/OR BLOWN FUSE
1X RPM
2X RPM
F_L
2F_L
3F_L
4F_L
5F_L
SCR FREQ.
E. FAULTY COMPARITOR CARD
1X RPM
2X RPM
3200 LINE SPECTRUM
SIDEBANDS LIKELY EQUAL TO SPEED VARIATIONS
SCR FREQ.
F. ELECTRICAL CURRENT PASSAGE THRU DC MOTOR BEARINGS
1X RPM
2X RPM
1600 LINE SPECTRUM
DIFFERENCE FREQUENCIES NORMALLY EQUAL BPFO IF FLUTING IS PRESENT
180K CPM



BELT DRIVE PROBLEMS
A. WORN, LOOSE OR MISMATCHED BELTS
VERT.
PERPENDICULAR TO BELT TENSION
PULLEY #1
HORIZ.
VERT.
BELT TENSION
PARALLEL TO BELT TENSION
PULLEY #2
HORIZ.
BELT FREQUENCY HARMONICS
1X DRIVEN
1X DRIVER
RADIAL IN LINE WITH BELTS
PITCH DIAM1 X RPM1 = PITCH DIAM2 X RPM2
B. BELT/PULLEY MISALIGNMENT
OFF SET
PIGEON TOE
ANGLE
AXIAL
1X DRIVER OR DRIVEN
C. ECCENTRIC PULLEYS
θ
RADIAL
1X RPM ECCENTRIC PULLEY
D. BELT RESONANCE
RADIAL
1X RPM
BELT RESONANCE

ΣΥΜΠΕΡΑΣΜΑΤΑ

Στο πλαίσιο της παρούσας πτυχιακής μελέτησα τις βλάβες που είναι πιθανό να εμφανιστούν σε μια μηχανή κατά τη λειτουργία της στη γραμμή παραγωγής καθώς επίσης και τις μεθόδους διάγνωσης αυτών και τη συντήρηση των μηχανών για την αποφυγή τους. Αυτό με βοήθησε στην εξοικείωση με την Αγγλική γλώσσα και ειδικότερα με την τεχνική ορολογία. Η διαδικασία αυτή αποτέλεσε μια εποικοδομητική και αρκετά χρήσιμη αλλά και ενδιαφέρουσα ενασχόληση, όσον αφορά στο αντικείμενο της διάγνωσης βλαβών και συντήρησης των μηχανών, αλλά και στην απόδοση του βιβλίου από τα Ελληνικά στα Αγγλικά. Η εν λόγω διαδικασία ήταν αρκετά απαιτητική λόγω των ειδικών όρων, ωστόσο αποτέλεσε έναν καλό τρόπο εμπλουτισμού των γνώσεων μου.

Η μετάφραση είναι κυριολεκτική και έχει ακολουθηθεί οσο το δυνατόν καλύτερα η διάταξη και το ύφος του συγγραφέα σε μια προσπάθεια να αποδοθεί επακριβώς το επιστημονικό του περιεχόμενο.

Εν κατακλείδι, ευελπιστώ η παρούσα διπλωματική να φανεί χρήσιμη στους συναδέλφους μηχανολόγους μηχανικούς αλλά και στους δυνητικούς φοιτητές ERASMUS, για τους οποίους θα αποτελέσει βοηθητικό μέσο για την καλύτερη κατανόηση του αντικειμένου.

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