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ΜΠΟΚΙΑΣ ΚΩΝΣΤΑΝΤΙΝΟΣ

ΠΟΛΥΚΑΝΔΡΙΩΤΗΣ ΒΑΣΙΛΗΣ

ΦΕΒΡΟΥΑΡΙΟΣ 2023

ΠΡΟΛΟΓΟΣ

Το παρόν τεύχος αποτελεί την Διπλωματική Εργασία που εκπονήθηκε στο Τμήμα Μηχανολόγων Μηχανικών του Πανεπιστημίου Πελοποννήσου και αναφέρεται στην μετάφραση του συγγράμματος "Μηχανές Εσωτερικής Καύσης", από τα Ελληνικά στα Αγγλικά, του διδάσκοντος και συγγραφέα του συγγράμματος στην ελληνική γλώσσα, Δρ. Κωνσταντίνου Μαυρίδη.

Η μετάφραση του συγκεκριμένου βιβλίου στα Αγγλικά, που είναι και διεθνής γλώσσα επικοινωνίας, έγινε για την εύκολη ανάγνωση του, από φοιτητές του τμήματος που προέρχονται από άλλες χώρες, μέσω του προγράμματος ERASMUS, αλλά και ημεδαπούς φοιτητές τους οποίους θα βοηθήσει εξαιρετικά κατά την διάρκεια των σπουδών τους και στην μετέπειτα πορεία τους στον χώρο της Μηχανολογίας.

Στόχος εκπόνησης αυτής της εργασίας είναι να μελετηθεί πολυδιάστατα η Μηχανή Εσωτερικής Καύσης δίνοντας βάση στην εφαρμογή και στην ανάλυση αυτής. Αυτό επιτυγχάνεται με την επιμέρους ανάλυση της ΜΕΚ από την ανάλυση κύκλων λειτουργίας και των συστημάτων ανάφλεξης έως και την έξοδο καυσαερίων. Αναλύεται σε επιμέρους σημεία όπως τη θερμική αλλά και η μηχανική ανάλυση της Μηχανής Εσωτερικής Καύσης. Εξετάζονται λεπτομερώς οι διαδρομές καυσίμου, αλλά και τα βασικά εξαρτήματα παλινδρομικής ΜΕΚ από τα οποία διέρχεται το καύσιμο.

Θα θέλαμε να ευχαριστήσουμε θερμά την επιβλέπουσα καθηγήτρια μας Κυρία Βασιλική Δούσμπη, του Τμήματος Μηχανολόγων Μηχανικών, για την πολύτιμη καθοδήγηση και βοήθεια, όπως και τον Κύριο Μαυρίδη Κωνσταντίνο που έδωσε άδεια να πραγματοποιηθεί η μετάφραση αυτού του συγγράμματος, ως μέρος της διπλωματικής εργασίας.

<u>Υπεύθυνη δήλωση φοιτητών:</u> Οι κάτωθι υπογεγραμμένοι φοιτητές έχουμε την επίγνωση των συνεπειών του Νόμου περί λογοκλοπής και δηλώνουμε υπεύθυνα ότι είμαστε συγγραφείς αυτής της Πτυχιακής Εργασίας, αναλαμβάνοντας επί ολοκλήρου του κειμένου εξ ίσου, έχουμε δε αναφέρει στην Βιβλιογραφία όλες μας τις πηγές τις οποίες χρησιμοποιήσαμε και λάβαμε ιδέες ή δεδομένα. Δηλώνουμε επίσης ότι, οποιοδήποτε στοιχείο ή κείμενο έχουμε ενσωματώσει στην εργασίας μας προερχόμενο από βιβλία ή άλλες εργασίες ή το διαδίκτυο, γραμμένο ακριβώς ή παραφρασμένο, το έχουμε πλήρων αναγνωρίσει ως πνευματικό έργο άλλου συγγραφέα και έχουμε αναφέρει ανελλιπώς το όνομα του και την πηγή προέλευσης.

Οι φοιτητές

(Ονοματεπώνυμο)	(Ονοματεπώνυμο)
Μπόνιας Κωνσταντίνος	Πολυανδριώτης Βασίλειος
………………	………………
(Υπογραφή)	(Υπογραφή)

ΠΕΡΙΛΗΨΗ

Η παρούσα Διπλωματική Εργασία αναφέρεται στην αρχή λειτουργίας των μηχανών εσωτερικής καύσης, την ιστορική τους αναδρομή και πώς ξεκίνησαν, στους κύκλους λειτουργίας των κινητήρων, τα εξαρτήματα τους και τα πλεονεκτήματα και μειονεκτήματα καθενός κινητήρα εξετάζοντας τον 4-χρονο, 2-χρόνο κινητήρα, και μιλά για τις ιδανικές συνθήκες λειτουργίας αυτών. Στη συνέχεια μας κατηγοριοποιεί τους κινητήρες ανάλογα με τον τρόπο λειτουργίας, αλλά και βάση του καύσιμου που δέχονται για να λειτουργήσουν και αναλύει την ισχύ, τον τρόπο υπολογισμού της αλλά και παράγοντες οι οποίοι επηρεάζουν την λειτουργία τους. Επίσης μας δείχνει τον τρόπο τον οποίο συνδέονται οι κατηγορίες ισχύος εξηγώντας μας τον βαθμό απόδοσης. Εξετάζει λεπτομερώς την καύση και τα συστήματα ανάφλεξης καθώς και τους τρόπους αποφυγής δυσλειτουργιών.

Στο πρώτο κεφάλαιο γίνεται εκτενής θερμική ανάλυση του κινητήρα και εξετάζονται σε κάθε κατηγορία οι κύκλοι λειτουργίας του εκάστοτε κινητήρα, ο θερμικός βαθμός απόδοσης και η ισχύς των διαφόρων κινητήρων και τα αίτια που επηρεάζουν αυτά τα μεγέθη και στην συνέχεια αναλύονται οι όροι βαθμός απόδοσης και ειδική κατανάλωση καυσίμου και η πίεση του εμβόλου και από τι αυτή αυξομειώνεται.

Στο δεύτερο κεφάλαιο παρουσιάζεται η καύση που πραγματοποιείται εντός του κινητήρα και ουσιαστικά μας δίνει την κίνηση των μηχανικών του μερών. Σε αυτό το σημείο παρουσιάζονται και αναλύονται ο τρόπος ανάφλεξης του καυσίμου σε κάθε κινητήρα και τα συστήματα ανάφλεξης, μαζί με τα εξαρτήματα και την αρχή λειτουργίας του κάθε είδους, αλλά και διάφορες δυσλειτουργίες που εμφανίζονται στην λειτουργία των κινητήρων λόγω ποιότητας καυσίμου και άλλων παραμέτρων.

Στο τρίτο στην σειρά κεφάλαιο γίνεται παρουσίαση του τρόπου με τον οποίο γίνεται ο σχηματισμός του καυσίμου μίγματος και πώς αυτός επηρεάζει τελικά την ποιότητα τις καύσης και τις εκπομπές αερίων, αλλά κα πώς αυτά επηρεάζουν το περιβάλλον. Ο κυριότερος διαχωρισμός που γίνεται σε αυτό το κεφάλαιο σχετικά με τον σχηματισμό του καύσιμου μίγματος και των εκπομπών είναι μεταξύ κινητήρων Όττο και Ντίζελ.

Στο τέταρτο κεφάλαιο εξετάζεται η μηχανική ανάλυση και η ζυγοστάθμιση των μηχανών εσωτερικής καύσης, και πιο συγκεκριμένα αναλύονται η κινηματική και η δυναμική τους, τα διαγράμματα ροπών και από ποιους παράγοντες εξαρτάται η τελική ροπή καθώς και τον τρόπο ζυγοστάθμισης ροπών και δυνάμεων.

Στο κεφάλαιο πέντε γίνεται αναλυτική καταγραφή των εξαρτημάτων μια παλινδρομικής μηχανής εσωτερικής καύσης και σε κάθε κεφάλαιο αναλύονται ξεχωριστά ένα, ένα τα εξαρτήματα της μηχανής τα οποία επιγραμματικά είναι: Έμβολο, ελατήριο εμβόλου, πείρος εμβόλου, διωστήρας, στροφαλοφόρος άξονας, σφόνδυλος, κύλινδρος και τέλος οι βαλβίδες και ο καθορισμός χρονισμού σε αυτές.

Στο έκτο και τελευταίο κεφάλαιο γίνεται αναφορά στην υπετροφοδότηση των κινητήρων με σκοπό να αυξήσουν τις επιδόσεις τους, αναφέροντας τρόπους με τους οποίους γίνεται αυτό σε κάθε είδους κινητήρα, τι αποτελέσματα έχει, αλλά και τι κινδύνους κρύβει.

Η Διπλωματική Εργασία αυτή μας βοήθησε στο να γνωρίσουμε και να εμβαθύνουμε σχετικά με την ιστορία, τον τρόπο λειτουργίας, τα είδη και τα πλεονεκτήματα και μειονεκτήματα που μας προσφέρει μια μηχανή εσωτερικής καύσης. Θωρούμε ότι οι μηχανές εσωτερικής καύσης είναι ένα μεγάλο μέρος της ανθρώπινης εξέλιξης, τις οποίες χρησιμοποιούμε καθημερινά στην ζωή μας, όπως για παράδειγμα στις μετακινήσεις μας με το αυτοκίνητο ή κάποιο άλλο μεταφορικό μέσο. Αν και τα τελευταία χρόνια υπάρχει μια στροφή προς τους ηλεκτρικούς κινητήρες, οι οποίοι δεν εκπέμπουν ρύπους στην ατμόσφαιρα και η πηγή ενέργειάς τους δεν είναι ορυκτά καύσιμα, αλλά ο ηλεκτρισμός που παράγεται και με πιο φιλικούς στο περιβάλλον τρόπους, σίγουρα ο άνθρωπος για πολλά χρόνια ακόμα θα εξαρτάται από τις μηχανές εσωτερικής καύσης.

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Internal Combustion Engines

SECOND EDITION

ION PUBLICATION

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ΕΙΣΑΓΩΓΗ

Οι μηχανές εσωτερικής καύσης είναι μια από τις σημαντικότερες εφευρέσεις για την τεχνολογική εξέλιξη του ανθρώπου καθώς δημιουργήθηκε η ανάγκη για πιο αποδοτικές, με χαμηλή κατανάλωση και μικρό όγκο μηχανές οι οποίες πλέον έχουν ευρεία χρήση ειδικά στην βιομηχανία των αυτοκινήτων και μοτοσυκλετών, άρα και των καθημερινών μας μετακινήσεων, όπως και σε πολλές ακόμα εφαρμογές. Οι Μηχανές Εσωτερικής Καύσης (Μ.Ε.Κ.) δημιουργήθηκαν για να αντικαταστήσουν τις δύσχρηστες, ογκώδεις, ακριβές και με μεγάλη κατανάλωση καυσίμου μηχανές ατμού που ήταν μέχρι τα τέλη του 1850 κυρίαρχο είδος μηχανής.

Οι Otto και Diesel πρωτοπόροι μηχανικοί της εποχής τους, από τους οποίους έχουν πάρει και έχουν κρατήσει μέχρι και σήμερα τα ονόματά τους οι αντίστοιχες μηχανές εσωτερικής καύσης, βλέποντας την ανάγκη που δημιουργήθηκε εκείνη την εποχή για πιο αποδοτικούς και εύχρηστους κινητήρες με χαμηλό κόστος, βασίστηκαν σε μια πρώιμη μηχανή εσωτερικής καύσης, του Lenoir (1822-1900) η οποία κατασκευάστηκε το 1860 και λειτουργούσε με φωταέριο ως καύσιμο. Η μηχανή αυτή είχε μεγάλη κατανάλωση και θόρυβο λειτουργίας, αλλά σχετικά καλή απόδοση και ήταν πιο εύχρηστη σε σχέση με τις ατμομηχανές. Εξέλιξαν βάση αυτής τις πρώτες μηχανές τους με παρόμοια αρχή λειτουργίας, οι οποίες σαφώς είχαν διαφορές μεταξύ τους, παρόλα αυτά σε μετεξέλιξη τους χρησιμοποιούνται ακόμα και σήμερα στην αυτοκινητοβιομηχανία.

Οι κινητήρες Otto επί το πλείστο χρησιμοποιούν την βενζίνη ως καύσιμο και έχουν χρήση σε αυτοκίνητα, βιομηχανία, μικρά πλοία και αεροσκάφη. Οι κινητήρες Diesel χρησιμοποιούν ως καύσιμο το πετρέλαιο και χρησιμοποιούνται στην αυτοκινητοβιομηχανία και ειδικά στα βαρέα οχήματα μεταφοράς, βιομηχανία, πλοία και στην παραγωγή ηλεκτρικής ενέργειας.

Πέρα από τον διαχωρισμό των μηχανών εσωτερικής καύσης ως προς το καύσιμο λειτουργίας τους, υπάρχει μια ακόμα βασική παράμετρος ανάλογα με τους χρόνους λειτουργίας. Υπάρχουν οι τετράχρονοι και οι δίχρονοι κινητήρες. Στους τετράχρονους κινητήρες για να πραγματοποιηθεί ένας ολόκληρος κύκλος λειτουργίας πρέπει το έμβολο να κάνει 4 διαδρομές. Αυτές είναι: 1) Εισαγωγή. Το καύσιμο εισέρχεται στο θάλαμο καύση από την ανοιχτή βαλβίδα εισαγωγής, 2) Συμπίεση. Το έμβολο κινείται προς το άνω νεκρό σημείο και συμπιέζει το μείγμα, 3) Ανάφλεξη. Το μείγμα καίγεται και εκτονώνεται, πιέζοντας το έμβολο στο κάτω νεκρό σημείο και παράγει έργο, 4) Εξαγωγή. Το έμβολο κινείται προς τα πάνω και σπρώχνει τα αέρια προς την ανοικτή βαλβίδα εξαγωγής διώχνοντάς τα έξω από τον θάλαμο καύσης. Στους δίχρονους κινητήρες γίνονται ακριβώς τα ίδια βήματα για την ολοκλήρωση ενός κύκλου λειτουργίας, απλά οι διαδρομές 1,2 εισαγωγή και συμπίεση γίνονται ταυτόχρονα και οι διαδρομές 3,4 εκτόνωση και εξαγωγή πάλι μαζί σε δεύτερο χρόνο.

Αν και τα τελευταία χρόνια υπάρχει μια τάση προς τον εξηλεκτρισμό των κινητήρων των αυτοκινήτων και των μοτοσυκλετών, εκεί όπου κατά κύριο λόγο χρησιμοποιούνται οι μηχανές εσωτερικής καύσης, οι αυτοκινητοβιομηχανίες δεν επιθυμούν να αποκοπούν από αυτούς τους κινητήρες τελείως καθώς έχουν επενδύσει τεράστια χρηματικά ποσά και χρόνο σε έρευνα με σκοπό την βελτίωση τους και την καλύτερη απόδοση. Σκοπός τους είναι με βάση τους κινητήρες αυτούς να κατασκευάσουν νέες παραλλαγές με παρόμοια αρχή λειτουργίας, οι οποίοι θα είναι πιο οικονομικοί, με μεγαλύτερη απόδοση και φιλικοί προς το περιβάλλον, καθώς γίνεται προσπάθεια ολοένα να μειώνονται οι εκπομπές καυσαερίων. Αποτέλεσμα αυτής της προσπάθειας είναι πολλές φορές η χρήση κινητήρα εσωτερικής καύσης με υποβοήθηση από ηλεκτρικό κινητήρα, έτσι ώστε όταν ανεβαίνει το φορτίο του κινητήρα ή αυξάνεται η επιτάχυνση με την βοήθεια του ηλεκτροκινητήρα παραμένει η λειτουργία του σε χαμηλές στροφές αποδίδοντας παράλληλα το απαραίτητο έργο, έχοντας μικρότερη κατανάλωση.

Για να γνωρίσουμε και να κατανοήσουμε τον τρόπο λειτουργίας των μηχανών εσωτερικής καύσης και να δούμε τα χαρακτηριστικά κάθε μίας και τις εφαρμογές της πρέπει να δούμε αναλυτικά τα κεφάλαια του συγγράμματος που ασχολείται και αυτή η εργασία.

Βασικές έννοιες, που θα μας απασχολήσουν στο πρώτο κεφάλαιο είναι η θερμική ανάλυση του κινητήρα που περιέχει τις ενότητες που αφορούν τους κύκλους λειτουργίας που διαχωρίζονται σε τετράχρονους κινητήρες και δίχρονους, την θερμική απόδοση και την ισχύ που παράγει και από ποιες παραμέτρους επηρεάζεται. Στις τελευταίες δύο ενότητες αναλύεται ο βαθμός απόδοσης, η ειδική κατανάλωση καυσίμου και η μέση πίεση εμβόλου.

Στο δεύτερο κεφάλαιο γίνεται αναφορά στην καύση και πιο συγκεκριμένα στα συστήματα ανάφλεξης τα οποία ανάλογα τον σχηματισμό τους στον θάλαμο καύσης, το μείγμα καυσίμου-αέρα και τον τρόπο ψεκασμού του καυσίμου έχουν διαφορετικά χαρακτηριστικά και πλεονεκτήματα ή μειονεκτήματα. Στο δεύτερο μέρος αναλύονται δυσλειτουργίες που εμφανίζονται στον κινητήρα, όπως ο χρονισμός ψεκασμού και χτυπήματα στην λειτουργία, από τα συστήματα ανάφλεξης ή το καύσιμο και τρόποι αντιμετώπισής τους. Στο τρίτο κεφάλαιο αναλύεται ο τρόπος σχηματισμού του καύσιμου μίγματος στου κινητήρες Otto και Diesel και του μεγέθους των ρύπων που εκπέμπουν ανάλογα με το καύσιμο και τρόποι που μπορούμε να μειώσουμε τις εκπομπές αυτές.

Στο τέταρτο κεφάλαιο, ο συγγραφέας επεξηγεί την μηχανική ανάλυση και ζυγοστάθμιση των κινητήρων αυτών. Στην πρώτη ενότητα παρουσιάζονται σε σχέδια και αναλύονται ο τρόπος κίνησης των μηχανικών μερών κάθε συστήματος και παρουσιάζονται κάποια από τα είδη μηχανής, όπως οι παλινδρομικές μηχανές όπου το έμβολο κινείται κάθετα πάνω κάτω ή περιστροφικές όπου το έμβολο κάνει περιστροφική κίνηση όπως στους κινητήρες Wankel, και πώς αυτά συνεργάζονται μεταξύ τους ώστε να δώσουν την ανάλογη δυναμική που χρειαζόμαστε σε έργο με σκοπό να αποδώσει περιστροφική κίνηση ο κινητήρας. Στη δεύτερη ενότητα δίνονται διαγράμματα και σχέδια σχετικά με το πώς επιδρούν οι ροπές επάνω στους κινητήρες και επιβαρύνουν τα μηχανικά τους μέρη και στο τέλος τρόποι ζυγοστάθμισης των δυνάμεων και ροπών έτσι ώστε να αποφευχθούν ανεπιθύμητες παρενέργειες από τριγμούς και ταλαντώσεις.

Στο πέμπτο και προτελευταίο κεφάλαιο γίνεται η ανάλυση, κάθε ενός ξεχωριστά των βασικών εξαρτημάτων μιας παλινδρομικής μηχανής εσωτερικής καύσης. Τα μέρη αυτά είναι: 1) Έμβολο, το οποίο κινείται μέσα στον κύλινδρο υπό μεγάλες μηχανικές και θερμικές πιέσεις και ουσιαστικά μετατρέπει την ενέργεια πίεσης που δέχεται στο επάνω μέρος της κεφαλής του σε μηχανικό έργο στο κάτω άκρο του που συνδέεται με τον άξονα, 2) Ελατήρια εμβόλου τα οποία έχουν 2 διαφορετικές λειτουργίες και είναι αυτά που στεγανοποιούν τον θάλαμο καύσης και αυτά που οδηγούν το λάδι λίπανσης στα τοιχώματα του κυλίνδρου, 3) Πείρος εμβόλου που μεταφέρει τις δυνάμεις στο κάτω μέρος μεταξύ του εμβόλου και του διωστήρα, 4) Διωστήρας, που συνδέει το έμβολο με τον στροφαλοφόρο άξονα, έχει κυκλική διατομή στον κορμό για καλύτερη λίπανση και ψύξη, ενώ έχει υποστεί θερμική επεξεργασία χάλυβα για σκλήρυνση, 5) Στροφαλοφόρος άξονας, μετατρέπει την παλινδρομική κίνηση της μηχανής σε περιστροφική κίνηση, 6) Σφόνδυλος, τοποθετείται στον κινητήρα ώστε να αποταμιεύει ενέργεια και να κρατάει τις στροφές λειτουργίας σταθερές καθώς η ροπή ενός εμβολοφόρου κινητήρα δεν είναι σταθερή λόγω της ίδιας της κίνησης που πραγματοποιεί,7) Κύλινδρος, είναι το εξάρτημα που περιβάλει και καθοδηγεί το έμβολο στην κατεύθυνσή του, 8) Οι βαλβίδες και ο καθορισμός χρονισμού τους, καθώς πρέπει να ανοίγουν για εισαγωγή φρέσκου αέρα και εξαγωγών καυσαερίων σε συγκεκριμένους χρόνους.

Στο έκτο και τελευταίο κεφάλαιο αναφέρεται στην υπετροφοδότηση κινητήρων, που συνήθως επιτυγχάνεται όταν το φρέσκο μίγμα αέρα οδηγείται στον κύλινδρο με υπέρ πίεση μέσω ενός συμπιεστή. Στην συνέχεια αναφέρονται τα πλεονεκτήματα αυτής της μεθόδου και τρόποι που μπορεί να γίνει, αλλά και προβλήματα που προκαλεί στην λειτουργία.

Introduction

The internal combustion engine is one of the most intrusive mechanical devices for humans, due to the highly competitive nature of the car market (in which most of these engines are used) and the emergence over time of major new problems.

Initially, the first attempts were made to increase the lifetime of internal combustion engines between two overhauls. Subsequently, the problems of air pollution were intensified and to this day manufacturers are making intensive efforts to improve engines in this area. The problem of noise reduction is also a real one, more serious for diesel engines. The large increase in fuel costs has recently led to the attention of car manufacturers to the need to improve fuel economy and to change the traditional engine design. Remarkably, under pressure to reduce fuel costs, the thermal efficiency of large marine diesel engines can nowadays exceed 50%. Efforts are also concentrated on the development of the so-called 'adiabatic engine' using ceramic parts and operating at very high temperatures, with the aim of increasing the thermal efficiency more and more. The trend towards the deterioration of available fuel resources is also leading to constant efforts to maintain efficiency by using poor-quality fuel.

For about a century two great names, Otto, and Diesel, played a major role in the rapid growth of the Internal Combustion Engine. In the past, the piston steam engine was the only thermal engine that could deliver mechanical power to man, but it was difficult to use because of the accessories (heater - boiler) and the lack of skilled craftsmen. Lenoir (1822-1900) built a gas engine in 1860, based on the piston steam engine, which was like a two-stroke, dual-energy engine. The piston absorbed the combustible mixture (photon gas - air) during half of its travel. The mixture was then electrically ignited, and the hot exhaust gases propelled the piston to the end of its stroke. On its return the piston expelled the exhaust gases while on the other side a new operating cycle was started. Lenoir's engine was water-cooled, had a very high-power consumption, and produced about 0.4-2.2 kW of steam with a gas consumption of $4m^3/kWh$. Its operating noise was high and harsh, but it was easier to set up and operate than the steam locomotive and for this reason it became a sought-after engine. Nicolaus August Otto (1832-1891) initially worked on improving Lenoir's engine and built a gas engine that operated under atmospheric conditions. Otto, in collaboration with the engineer Eugen Langen (1835-1895), reduced the noise from combustion strokes and greatly reduced the consumption of flue gas compared to Lenoir's engine. The power of the first atmospheric flue gas engines was about 0.7kW and their height was up to 2m; higher power engines had a very high construction height and presented many space installation problems. The demand for higher power engines prompted Otto to design a new type of engine , with a direct crankshaft piston connection and application for the first time of the four-stroke operating process: a) Aspiration of the flue gas-air mixture, b) Compression of the mixture, c) Combustion of the gas mixture, d) Exhaust gas extraction. The revolutionary point of the new engine was the four-stroke cycle and the first ever engine in which we have mixture compression before combustion. The first four-stroke engine was built in 1876 and produced 2.2 kW at 180rpm. This engine is the basis of all four-stroke engines today. A few years later, a new engine, the Diesel engine, was built by Rudolf Diesel (1858-1913). Running engines on high-temperature steam gave him the idea to build an engine that would run on high-temperature supercharged air. After painstaking efforts and improvements , in 1897 Diesel was able to present his engine in its final form, which produced 13.1kW at 154rpm. Fuel consumption was 324gr/kWh. With this low consumption, the diesel engine surpassed in terms of economy all the thermodynamic engines known up to that time, an advantage it still retains today.

Today, it is common practice to describe internal combustion engines with external ignition as Otto engines and with compression ignition as diesel engines. Thermal engines can therefore be classified as external combustion engines and internal combustion engines. In the former, the operating fluid is internally separated from the fuel-air mixture and the heat of combustion is transferred through the walls of the combustion vessel or boiler. In internal combustion engines the fluid content consists of the combustion products of the same fuel-air mixture. The following table gives a classification of the most important types of thermal engines and their fields of application.

ΚΑΤΑΤΑΞΗ ΘΕΡΜΙΚΩΝ ΜΗΧΑΝΩΝ

ΚΑΤΗΓΟΡΙΑ	ΟΝΟΜΑ	(1) ΠΑΛΙΝΔΡΟΜΙΚΗ (2) ΣΤΡΟΒΙΛΟΣ	ΧΡΗΣΗ	ΚΑΤΑΣΤΑΣΗ
ΕΞΩΤΕΡΙΚΗΣ ΚΑΥΣΗΣ	ατμομηχανή	(1)	οχήματα	απηρχειωμένη
	ατμοστρόβιλος μηχανή	(2)	ηλεκτρική ενέργεια πλοία	σε ενέργεια
	θερμού αέρα	(1)	καμία	απηρχειωμένη
	αεριοστρόβιλος κλειστού κύκλου	(2)	ηλεκτρική ενέργεια πλοία	σε πειραματικό στάδιο
ΕΣΩΤΕΡΙΚΗΣ ΚΑΥΣΗΣ	βενζινομηχανές	(1)	αυτοκίνητα μικρά πλοία αεροπλάνα μικρές βιομηχανές	σε ενέργεια
	μηχανές DIESEL	(1)	αυτ. βιομ. οχήματα πλοία ηλ. ενέργεια	σε ενέργεια
	αεριομηχανές	(1)	βιομ. ηλ. ενέργ.	σε ενέργεια
	αεριοστρόβιλοι	(2)	ηλ. ενέργεια αεροπλάνα	σε ενέργεια
	μηχανές JET	(2)	αεροσκάφη	σε ενέργεια

Today reciprocating internal combustion engines and steam turbines are for the most part the most used internal combustion engines, with the gas turbine in widespread use in the propulsion of high-speed aircraft. A fundamental advantage of reciprocating internal combustion engines over other types of engines is the absence of thermal exchangers in the flow of the operating fluid, such as boilers and condensers in steam power plants. The absence of the elements not only leads to mechanical simplification but also alters the existing losses in the heat transfer process in the exchangers. The reciprocating internal combustion engine has another important and fundamental advantage over the gas turbine: All its components can operate at temperatures well below the maximum cycle temperature. This allows the very high cycle temperatures to be used and thus high cycle efficiencies to be possible. In design, these key differences give reciprocating internal combustion engines the following advantages in terms of power output when compared with steam turbines: 1) Higher maximum efficiency, 2) Lower weight ratio of construction to power output (except in the case of units above 10000hp, 3) Mechanical simplicity, 4) The cooling system of an internal combustion engine experiences a smaller amount of heat than a steam turbine condenser of the same horsepower and operates at higher surface temperatures. The smaller size of the heat exchanger is a great advantage in transport vehicles and other applications where cooling must be supplemented by atmospheric air. These advantages are partly evident in small units. In contrast, practical advantages of turbine engines over reciprocating engines are: 1) Turbine engines can use a wider variety of fuels including solid fuels, 2) Less vibration problem, 3) Steam turbines are practical for very large power units (over 200000hp).

The advantages of reciprocating internal combustion engines are considered of great importance in the field of land transport, where small weights, small engine volume and small amount of fuel are usually essential factors. In today's world, the number of units and the total power of internal combustion engines in use is much greater than all other means of transport. Also, the internal combustion turbine engine has not been fully established as a competitive engine in the power generation industry outside of aircraft. The mechanical simplicity of this engine makes it very interesting, and the absence of reciprocating components eliminates vibration in proportion to the steam turbine.

CHAPTER 1

1 Engine Thermal Analysis

1.1 Operating Cycles

The duty cycle is the process by which the energy input to the engine from the fuel is converted into mechanical work. Two operating cycles are distinguished:

the two-stroke and the four-stroke duty cycle. Both diesel and Otto engines operate according to these cycles. Today almost all Internal Combustion Engines are single-acting, meaning only one side of the piston is in contact with the combustion gases.

1.1.1 Four stroke Operation Cycle

A four-stroke operating cycle is carried out with four piston strokes or two rotations of the crankshaft.

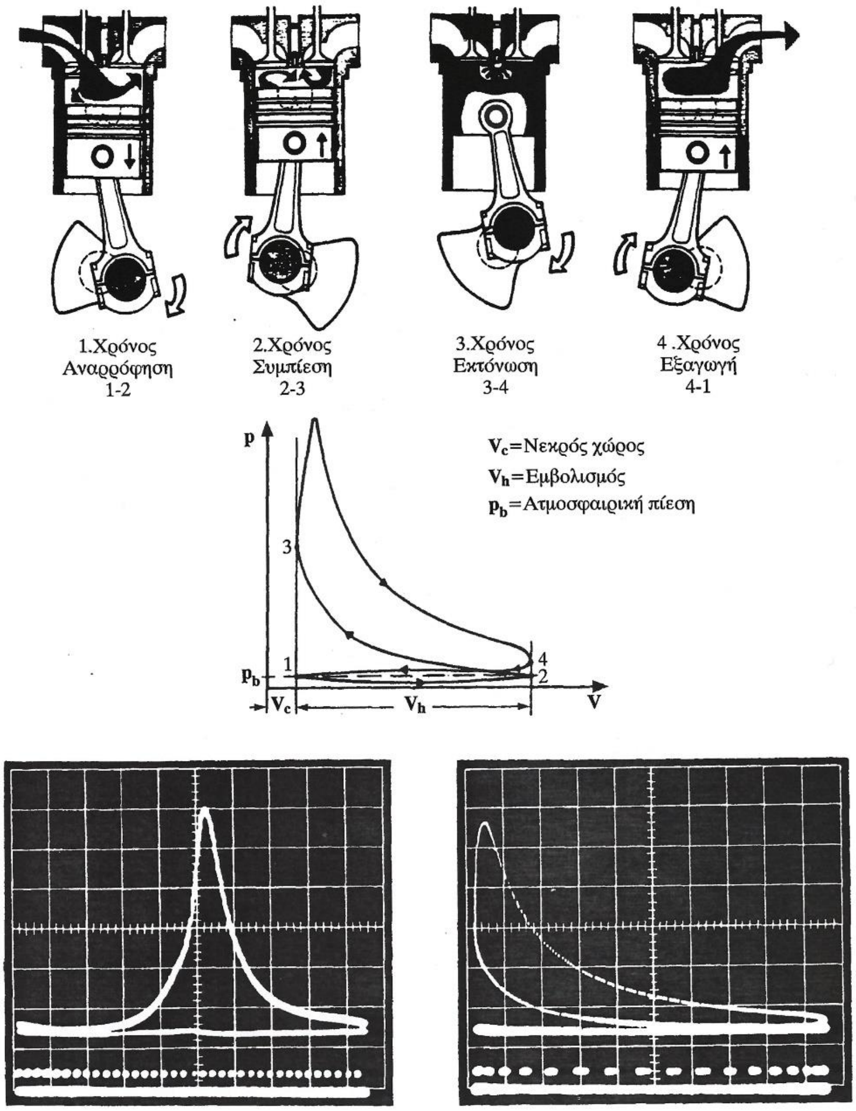
In the first stroke (suction) the piston moves from top dead center to bottom dead center with the intake valve open and the exhaust valve closed and simultaneously sucks fresh mixture into the cylinder. Currently the cylinder is under pressure by a few tenths of a bar.

In the second stroke (compression), the piston is driven from the bottom dead center to the top dead center with the valves closed and compresses the mixture. The pressure and temperature increase. The final values range from 30 to 50 bar and 550 to 700 degrees Celsius for oil engines and from 10 to 16 bar and 350 to 450 degrees Celsius for the Otto engine.

In the third stroke (work time) the valves are closed. Fuel combustion starts when the piston is approximately at top dead center. This causes the temperature and pressure to rise and reach the highest values (in the diesel engine 2000 degrees Celsius and 60 to 100 bars and in the Otto engine about 2500 degrees Celsius and 40 to 70 bars). After combustion the gases expand and only during this time work is transferred from the gases to the piston. During the other three cycles, the piston delivers work to the gases.

In the fourth stroke (exhaust), with the exhaust valve open and the intake valve closed, the piston expels the exhaust gases from the cylinder. The cylinder is slightly pressurized.

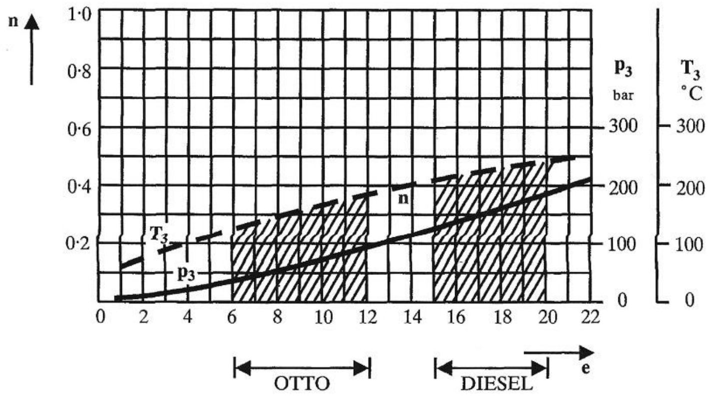
In drawing (1.1.1.1.1) the four times of the engine and the corresponding p-V, p-Θ indicative diagrams of a four-stroke engine are given.



Drawing (1.1.1.1.1) four-stroke cycle and p-V, p-Θ diagrams.

In a diesel engine at 3000 rpm the available time for fuel injection, ignition and combustion is two milliseconds and the air temperature rise is about 1000 degrees Celsius. In the petrol engine, which can very easily rotate at three times the speed, we have available time of much less than one millisecond for the entire combustion process. Of course, it took a lot of effort to achieve the above results.

Fuel is burnt in the engine for one reason only, to raise the air temperature. In the petrol engine we compress the filling air to a range of pressures from 9 bar (small, air-cooled engine) to 25 bar (typical modern car engine) and even 35 bar (high performance racing car engine) corresponding to compression ratios of 5:1, 10:1 and 14:1 with corresponding air temperatures of about 300, 450 and 550 degrees Celsius (drawing 1.1.1.2).



Drawing (1.1.1.2) Otto-Diesel engine compression limits

At the end of the compression stroke, (drawing 1.1.1.1), the fuel is burned to raise the air temperature, increase the pressure, get more work from the engine at the time of expansion. By burning the entire amount of oxygen in the air, the temperature rises to 2600 to 3000 degrees Celsius, and this is a limit to the increase in pressure and therefore power of engine.

It should be emphasized that an internal combustion engine is a gas engine, and its efficiency is determined by the properties of the air, not the fuel. It is easy to introduce into the engine as much fuel as we want, but it is incomparably difficult to fill it with clean air, and it is the amount of air drawn into the engine that determines its power.

In a standard petrol engine, with a carburetor and no injection system, the air on its way to the engine also draws liquid petrol from the carburetor. Despite all efforts to evaporate the gasoline, it enters through the intake as a rapid rain. During compression, it is the increase in air temperature and contact with the hot surfaces of the cylinder piston and combustion chamber that ensure that the petrol is condensed. Shortly before the end of the compression stroke, we have an explosive mixture with a temperature of about 400 degrees Celsius, which we seek to burn smoothly and quickly. This is accomplished by sparking at about 10kV at the tip of the spark plug, at an angle of ten to thirty degrees to the crankshaft before the engine's top dead center depending on the engine rpm. The flame begins to grow at the tip of the scintillator, becomes a spherical front of fire that moves rapidly sweeping across the combustion chamber and the whole process is completed thirty degrees past top dead center, where the flame is extinguished by reaching maximum temperature.

The time delay between the ignition delay from the silence of the spark and the change from slow to rapid spread of the flame is uncontrollably variable. A critical factor is the rate of pressure rise in the combustion chamber , which depends on the speed at which the flame front moves. If the flame front is moving too fast and the pressure rises sharply, noisy, and erratic operation of the engine ensues. Conversely, if combustion is completed with a delay, there is a loss of power at the time of expansion. Control of the rate of flame spread is one of the main problems of petrol engines. The main control factors are the shape of the combustion chamber, the position of the scintillator and the intensity of the turbulence. In the petrol engine, a confined combustion chamber is sought so that the flame is not widely spaced, the scintillator is placed at the hottest point of the combustion chamber (above the exhaust valve) and the turbulence intensity is the desired intensity for the internal combustion engine.

In the petrol engine there are two undesirable situations which are pre-ignition and knocking. Pre-ignition occurs when the fuel mixture ignites before a spark is given and occurs either at the point of the exhaust valve or because of unburnt carbon deposited on the walls of a dirty combustion chamber. The latter cause can cause the engine to continue to rotate even when the switch is turned off. Even more troublesome is the "knock" that results from running the engine on poor quality fuel, poor premature spark advance, or running the engine at low rpm at full power. Engine knock occurs when the flame front ceases to develop smoothly in the combustion chamber and the entire unburned mixture explodes instantaneously.

Increasing the degree of turbulence to make the flame reach all parts of the combustion chamber more quickly may make the engine run erratically due to the sudden increase in pressure. With a low degree of turbulence , the flame moves more slowly, but then the mixture is given enough time available to explode uncontrollably and a hit occurs. Measures that can be taken are to place the scintillator at the hottest point (on the exhaust valve) and to design the combustion chamber in such a way that the last amount of fuel mixture to be burned is confined to the coolest part of the chamber (part cooled by the water-cooled walls of the chamber). The early engines could not operate with compression ratios above 4:1 because they did not work properly, the flame path was too long, and combustion started in the cooler area and ended in the warmer area near the exhaust valve.

Another factor of fuel quality, namely its resistance to pre-ignition, is expressed by the octane number. Nowadays, it is possible by using additives to achieve octane numbers higher than one hundred, but there is a tendency not to use fuels with very high-octane numbers because of the high cost and to avoid air pollution. The use of high-octane petrol increased the compression ratio from 6:1 forty years ago to 9:1 in 1960, but there has been no further increase since then. The ideal thermal efficiency is about 36% for a 4:1 compression ratio, which increases to 50% for a 10:1 compression ratio (the thermal efficiency of a real engine is 2/3 of the ideal).

In a standard gasoline engine, a very rich mixture corresponds to an air/fuel ratio of 7:1 (enough air to burn only half the fuel), while a very poor one corresponds to a ratio of 22:1 (enough fuel to use only 2/3 of the oxygen in the air). Measurements of output power and specific fuel consumption by varying the air/fuel ratio from the correct air/fuel ratio of 15:1 provide the following conclusions. IF the mixture becomes richer than the correct ratio, power increases slightly by 4% (maximum power output at the 13:1 ratio), but specific fuel consumption increases faster. Continued enrichment of the mixture leads to a decrease in power, while the engine starts to run erratically, and explosions can occur in the exhaust and muffler. Conversely , with a poorer mixture than the correct ratio, the power starts to decrease but the specific consumption improves and at a power output of 85% of the original power the specific consumption decreases by about 4% and the air ratio is about 17:1. An even poorer mixture leads to an increase in fuel consumption and a decrease in power, while the engine operation is determined to be unstable with a risk of explosions in the exhaust and also engine stoppages occur.

1.1.2 Two stroke Operation Cycle

The two-stroke cycle is carried out with only two pistons or one rotation of the crankshaft. The two times of suction and exhaust are not saved, but are replaced by the operation of another mechanism, the sweep compressor. In the two-stroke Otto engine , the underside of the piston works as a sweep compressor. Because the diesel engine needs more air than the lower side of the piston can supply, a special compressor is built (Figure (1.1.2.1)).

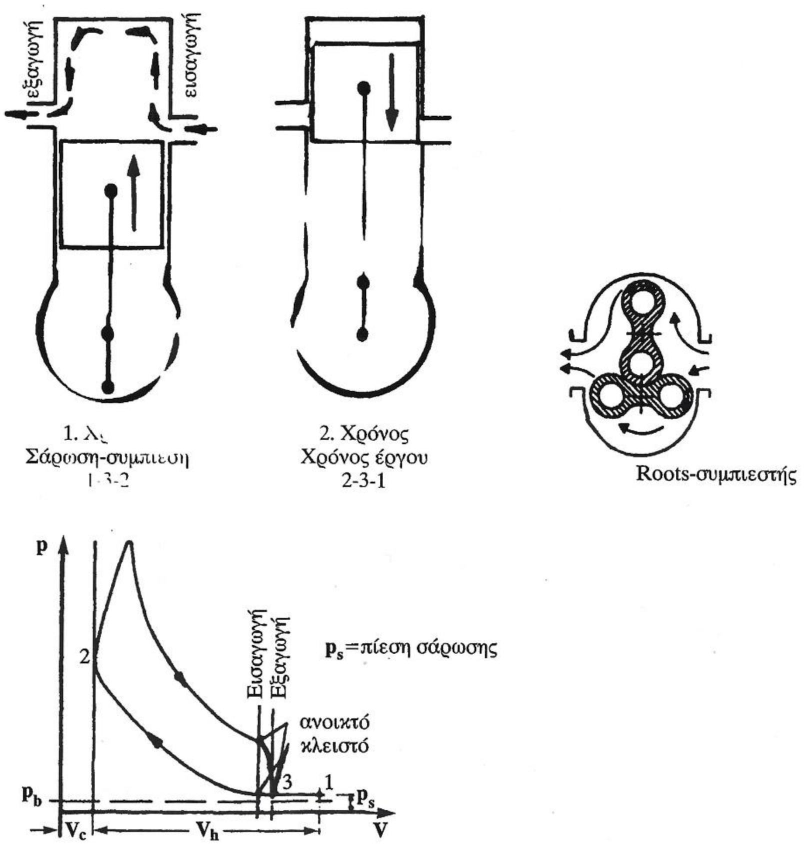


Figure (1.1.2.1) Two-stroke engine cycle and air compressor

The operating cycle of the two-stroke machine is carried out in 2 times: In the first time (sweep, compression), the piston moves from bottom dead center to top dead center and before covering the intake and exhaust ports, the fresh mixture sweeps the combustion gases from the cylinder.

To achieve a better flushing of the cylinder, the fresh mixture is compressed in the sweep compressor and at a pressure slightly higher than that of the exhaust gases. After the ports are closed by the piston, the fresh mixture is compressed. The two-stroke engine has values of viscosity and calorific values corresponding to those of the four-stroke engine.

In the second time (work time), combustion begins as in the four-stroke engine, when the piston is at about top dead center. The temperature and pressure reach approximately the same values as in the four-stroke engine. The operating medium is then discharged; when the piston releases the exhaust port, the exhaust gases are forced out towards the exhaust. Soon after, the intake port opens and the fresh mixture that flows in sweeps (flushes) the exhaust gases from the cylinder to the exhaust. Figure (1.1.3) shows the two times of the two-stroke engine and the corresponding p-V diagram.

1.1.3 Ideal Operating Conditions (Processes)

Ideal operating conditions (processes) are the ideal cyclic processes under which an ideal machine could operate. With the aid of such processes, engines with different operating cycles can be compared with each other with reference to their economy. The actual operating processes usually deviate significantly from the ideal processes and their economic efficiency is much lower. Changes in its course can be more easily examined from a computational point of view under ideal conditions and their effect on the economics of the actual process can be assessed. Therefore, ideal processes are an important means for selecting the work path in the real machine. For the Otto engine the isochronous process was chosen as the ideal process and for the diesel engine the mixed process (Figure1.1.3.1)).

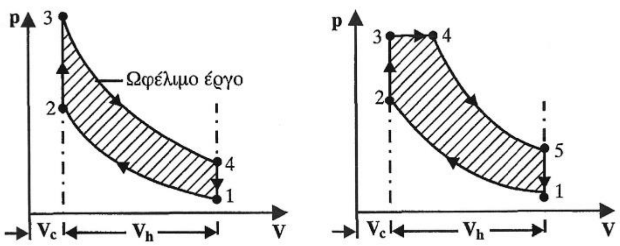


Figure (1.1.3.1) Ideal Otto - Diesel cycles

The mixed ideal process is the ideal process in the diesel engine. For this process, it is assumed that a portion of the fuel is burned isochronously (instantaneously) and the remainder is fed in such a way that combustion takes place under constant pressure. The following state changes occur in the cyclic process: 1-2 isentropic compression, 2-3 heat supply under constant pressure, isothermal combustion, 3-4 heat supply under constant pressure, isothermal combustion, 4-5 isentropic expansion and 5-1 heat removal under constant pressure.

The isochoric process is the ideal Otto engine process. In this process the total fuel is burned instantaneously, under constant pressure. The cyclic process consists of the following state changes: 1-2 isentropic compression, 2-3 heat supply under constant pressure, isochronous combustion, 3-4 isentropic expansion, 4-1 heat removal under constant pressure.

The comparison of the two processes shows that the isochoric process is a special case of the mixed ideal process. It is observed that if the isothermal combustion is zeroed, the 4 point is transferred to 3, then the mixed process switches to the isochoric process.

In an ideal engine, all quantities are required to be the same as in the real engine. Only the pure air-fuel mixture and no residual exhaust gas from the previous operating cycle must be present in the cylinder. The air ratio (λ) must be equal to that of the actual engine. The fuel must be fully combusted. There must be no heat exchange between the operating medium and the walls surrounding it. There must be no leakage during the inflow and outflow of gases. During the operating cycle it is advisable that no gas leaks occur due to poor sealing. The operating medium is ideal and not real gas. The latter means that the specific heat capacity does not vary with temperature and that at high temperatures no molecular decomposition occurs.

The calculation of the cyclic process of an ideal machine, which satisfies the previous conditions, gives good ideal values, but is particularly complicated because of the last condition. By considering the ideal gas a considerable simplification of the calculation is obtained, but on the other hand the efficiency becomes very high.

The actual operating cycle of the engine differs considerably from the ideal one. In particular, the following differences can be observed: Not only clean mixture and the residual exhaust gas from the previous operating cycle are present in the cylinder. The fuel is not completely burnt. The combustion is neither under constant volume nor under constant pressure.

The gas exchanges heat with the walls. During the inflow and outflow leaks occur. The gases escape through the piston springs.

The development of the actual operating process is measured by the indicator, which records the pressure in the cylinder in relation to the piston stroke or time. Deviations from the operating cycle of the ideal machine can be seen in the illustrative diagram.

Figure (1.1.3.2) shows an illustrative diagram together with the diagram of the isochronous process. The flow losses during the inflow and outflow process, the heat exchange from the boundaries and the variations during the combustion process are evident.

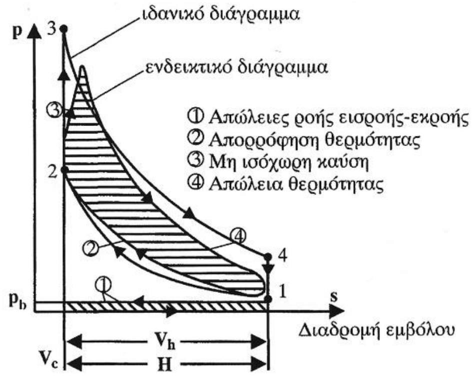


Figure (1.1.3.2.) Illustrative and ideal diagram of a four-stroke Otto engine.

1.2 Thermal efficiency

The thermal efficiency is the ratio of the useful power to the power of the ideal engine.

$$\mathrm{nt}h = \frac{L}{Q} \tag{1.2.1}$$

$$L = Q - QE \tag{1.2.2}$$

$$\mathrm{nth} = 1 - \frac{QE}{Q} \tag{1.2.3}$$

L= useful heat in unit time, Q= supplied heat in unit time, QE= dissipated heat in unit time.

The previous relationship is used to calculate the thermal efficiency. This formula is not suitable for calculating the efficiency based on the engine data.

Below are the relationships for determining the efficiency , first for the mixed process and then for the isochronous process. The heat content per operating cycle consists of two individual heat contents. Input heat under constant volume V and input heat under constant pressure p.

$$Q1 = mcv(T3 - T2) = mcvT2\left(\frac{\mathrm{T3}}{T2} - 1\right) \tag{1.2.4}$$

$$\mathrm{Q2 = mcp(T4 - T3) = mcpT3}\left(\frac{T4}{T3} - 1\right) \tag{1.2.5}$$

m = mass of gas per cycle and cv, cp = specific heat capacities. To simplify the calculations, the temperature dependence is not considered, as well as the decomposition of molecules due to high temperature. By introducing the isotropic (multimodal) exponent κ=cp/cv :

$$\mathrm{Q = Q1 + Q2 = mcv}\left\{\left[\frac{\mathrm{T3}}{\mathrm{T2}} - 1\right]\mathrm{T2} + \kappa\left[\frac{T4}{T3} - 1\right]\mathrm{T3}\right\} \tag{1.2.6}$$

The corresponding quantities are used:

compression ratio $\varepsilon = \frac{Vh+Vc}{Vc} = \frac{V1}{V2},$

pressure ratio $\beta = \frac{p3}{P2} = \frac{T3}{T2},$

injection ratio $\gamma = \frac{V4}{V3} = \frac{T4}{T3}$

For the isotropic change from 1 to 2, the following applies:

$$T2 = T1 * \varepsilon^{k-1}$$

With the help of this relationship the heat supplied per operating cycle is:

$$Q = mcv * T1 * \varepsilon^{K-1}(\beta - 1 + \kappa\beta * (\gamma - 1) \tag{1.2.7}$$

The heat dissipated per cycle is:

$$QE \ = mcv * T1 * (\gamma^{\kappa} * \beta - 1) \tag{1.2.8}$$

The formula for the thermal efficiency of the mixed ideal process is derived from the previous equations. In relation (1.2.3) the heat fluxes are replaced by the heat quantities per operating cycle.

$$nth = 1 - \frac{1}{\varepsilon^{\kappa-1}} * \frac{\gamma^{\kappa} * \beta - 1}{\beta - 1 + \kappa\beta(\gamma - 1)} \tag{1.2.9}$$

With this relationship the efficiency of the ideal diesel engine can be calculated, because ε and β are known and γ can be determined from other data quantities. The thermal efficiency of an isothermal process is obtained through relation (1.2.9) in which γ = 1. Then in the ideal process, point 4 changes to 3. The mixed ideal process is converted into an isochronous process. The efficiency of the ideal motor is given by the relation.

$$nth = 1 - \frac{1}{\varepsilon^{\kappa-1}} \tag{1.2.10}$$

With the same compression ratio e the isochronous process has the best efficiency, because the expression:

$$\frac{\gamma^{k}\ \beta - 1}{\beta - 1 + \kappa\beta(\gamma - 1)} > 1$$

At higher compression ε, both efficiencies increase. The compression ratio is approximately 16 to 24 for diesel engines and 7 to 10 for Otto engines. The above figures show that the Diesel engine has a better efficiency when running an equivalent indicative diagram to the Otto engine, although its cyclic process is less productive than that of the Otto.

1.3 Power

Different expressions of power are distinguished in engines, depending on the location and conditions of its determination in the engine.

1.3.1 Internal Power

The internal power Pi is called indicative power because it is determined by the indicative diagram. This power is transferred from the actuating medium to the piston. The internal power is calculated with the aid of the average piston pressure as follows:

$$Pi = pi * Vh * zni \tag{1.3.1.1}$$

z = number of cylinders, n = rpm, i = 0,5 in the case of a four-stroke engine.

1.3.2 Effective (active) power

The effective or active power Pe is the power delivered to the clutch (axle) of an engine. This power is smaller than the internal power by the power of friction.

1.3.3 Friction power

The friction power consists of the friction power in the piston, piston springs and other components of the engine drive mechanism, the power required to start any necessary auxiliary devices such as the petrol pump, water pump, oil pump, fan, injection pump, sweep compressor and dynamo (at idle). The net power is measured with the power measuring pens.

$$P\tau\rho = Pi - pE \tag{1.3.3.1}$$

Pτρ = friction power internal (indicative) power Pε= useful (active) power. Instead of the above relationship, the corresponding relationship for average pressures is often used.

$$p\tau\rho = pi - p\varepsilon \tag{1.3.3.2}$$

pτρ= average friction pressure, pi= average piston pressure, pε= average effective pressure. In relation 1.3.3.2) by dividing by the product Vhzni, the relation for average pressures (1.3.3.2) is obtained.

The friction force can be expressed in terms of the mechanical efficiency nm.

$$nm = \frac{P\varepsilon}{Pi} = \frac{P\varepsilon}{P\varepsilon + P\tau\rho} \qquad (1.3.3.3)$$

Solving the above relation in terms of the friction power Pτρ:

$$P\tau\rho = P\varepsilon\left(\frac{1}{nm} - 1\right) \tag{1.3.3.4}$$

Pτρ = friction power, Pε = effective power, nm = mechanical efficiency. From a technical and scientific point of view, it is desirable that the friction power, which is a loss power, be kept as low as possible. Its exact determination is far from simple. The exact method applies the relation (1.3.3.1). This also means that the internal power and the useful power must be determined very precisely. Small inaccuracies result in a large calculation error, because the power of friction is a small difference in the value of two large quantities.

Because the exact determination of the internal force is quite difficult, approximate methods of calculation are often applied.

1.3.3.1 Determination of the friction power by means of fuel consumption.

According to this method, the engine torque is varied by keeping the engine speed constant and at the same time the fuel consumption is determined. The torque is used to calculate the average effective pressure and the fuel consumption is used to calculate the amount of fuel consumed, as shown in an illustrative diagram. From the relationships:

$$P\varepsilon = p\varepsilon * Vh * zni \tag{1.3.3.1.1}$$

$$P\varepsilon = M * 2 * \pi n \tag{1.3.3.1.2}$$

Pε = net power, ρε = average net pressure, Vh = cylinder swell, z = number of cylinders, N = number of revolutions, i = indicative number of diagrams per revolution, M = engine torque. From the above 2 relations if we divide them by parts and solve for the average effective pressure ρε:

$$p\varepsilon = \frac{2\pi M}{Vhzi} \tag{1.3.3.1.3}$$

pε=average useful power, M=torque, Vh=cylinder engagement, z=number of cylinders, i=number of indicative diagrams per revolution.

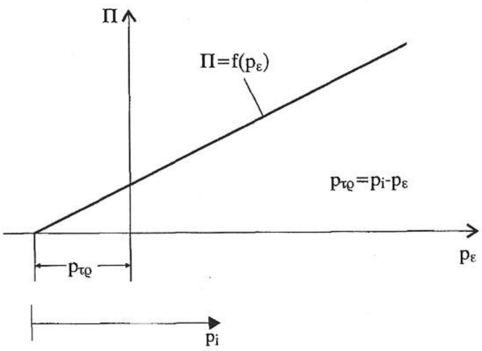


Figure (1.3.3.1.1) Calculation of average friction pressure using the amount of fuel per indicator diagram.

The relationship for the quantity of fuel per indicator diagram is:

$$\Pi = \frac{K}{zni} \tag{1.3.3.1.4}$$

Π = fuel quantity per indicator diagram, K = fuel consumption, n = number of revolutions, z = number of cylinders, i = number of indicator diagrams per revolution.

Plot the diagram of the fuel quantity P, per indicative diagram, against the average net pressure (Figure 1.3.3.1.1).

The curve Π= f(pε) does not start at the origin of the coordinates because the engine in idle mode (Pε=0) consumes fuel. The existing internal power corresponds to the friction power of the engine. To determine the friction pressure, the curve is extended to the left until it meets the intercept axis. At this point of intersection, the amount of fuel per indicator diagram and the average piston pressure is zero.

From the relationship ptp=pi-pε the difference between the average pressure on the piston and the average effective pressure is the average friction pressure. This corresponds to the distance between the point of intersection of the curve with the intersection axis and the beginning of the axes. The friction force calculated from the relationship:

$$P\tau\rho = P\tau\rho * Vh * zni \qquad \textit{(1.3.3.1.5)}$$

Pτρ = friction power, pτρ = average friction pressure, Vh = ramming of a cylinder, z = number of cylinders, n = number of revolutions, i = number of indicative diagrams per revolution.

The way of calculating the friction power described above is simple and correct; however, it does not take into account that the power friction also depends on the gas pressure in the cylinder and the temperature of the engine. In conditions of high pressure and low engine temperatures the magnitude of friction increases. The superior method should only be used for generalized friction power calculations.

1.3.3.2 Determination of the friction power by the method of shutdown.

The operating principle of this method is based on the fact that a rotating mass, by removing the kinetic force, is reduced in rotation and eventually brought to a standstill due to the frictional torque. From the moment of inertia of the mass and the angular deceleration, the frictional torque is calculated:

$$M\tau\rho = \Theta * \alpha \qquad (1.3.3.2.1)$$

Mτρ = friction torque, Θ = mass moment of inertia, α = angular deceleration.

The friction force is calculated from the relation:

$$P\tau\rho = M\tau\rho * \omega \qquad (1.3.3.2.2)$$

Pτρ = friction power, Mτρ = friction torque, ω = angular velocity.

In the next step the moment of inertia of all rotating masses and the effect of the reciprocating masses are determined (chapter 4.3). Then the engine stopping curve (figure (1.3.3.2.1)) is constructed according to the following methodology:

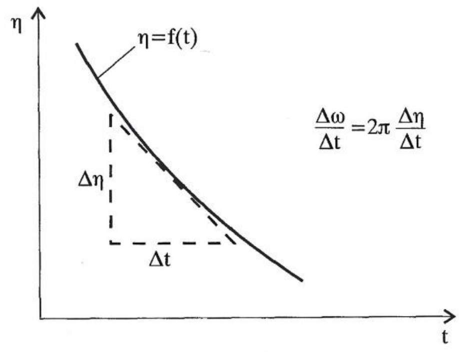


Figure (1.3.3.2.1) Engine stopping curve.

The engine is run at full power and then cut off. The diagram of the number of revolutions n, corresponding time t is drawn. The angular deceleration a is obtained if the tangent to the corresponding number of revolutions is brought to the curve n = f(t) in figure (1.3.3.2.1).

$$\alpha = \frac{\Delta\omega}{\Delta t} = 2\pi \frac{\Delta n}{\Delta t} \quad (1.3.3.2.3)$$

α = angular deceleration, ω = angular velocity, t = time

To the above method of determining the friction force, the error of calculating the mass moment of inertia and the error of the decoupling process of a cylinder, multi-cylinder engine is added.

1.3.3.3 Determination of friction power by decoupling a cylinder of a multicylinder engine.

The procedure followed according to this method is as follows:

First, the engine power verified when all cylinders are working. Then one cylinder is disconnected from operation and the power of the engine is determined again. From the measurement data we can calculate the friction power of a cylinder as follows:

A) Engine working with all cylinders. The power of P(z). The power per cylinder:

$$\frac{P(z)}{z} \quad (1.3.3.2.1)$$

P(z) = motor power z cylinders, z = number of cylinders.

B) Engine working with z-1 cylinders. Its power:

$$P(z-1) \quad (1.3.3.2.2)$$

C) The engine has z-1 cylinders so its power is

$$\frac{nP(z)}{z} * (z-1) \quad (1.3.3.2.3)$$

The power difference obtained from relations (1.3.3.2.3) and (1.3.3.2.2) is the friction power of the decoupled cylinder. From the above methodology we obtain the relation for calculating the friction power of one cylinder:

$$P\tau\rho = \frac{P(z)}{z} * (z-1) - P(z-1) \qquad (1.3.3.2.4)$$

Pτρ = single cylinder friction power, P(z) = motor power when all cylinders are working, z = number of cylinders, P(z-1) = motor power when one cylinder is disconnected.

The calculated friction power is less than the actual power because of temperature and the decoupling time error.

1.4 Efficiency and Specific Fuel Consumption

The efficiency describes the relationship between two powers. The thermal efficiency has already been mentioned. Apart from that there are other important efficiency factors for the evaluation of an engine. The efficiency is the ratio of the internal power to the power of the ideal engine.

$$\mathrm{ng} = \frac{Pi}{PI}$$

ng = quality grade, Pi = indicative power, PI = power of ideal engine.

This ratio indicates how high the quality of the actual engine is and how close the actual operation is to the ideal. Behavioral values:

Engine Otto ng = 0,4 to 0,7

Diesel engine ng = 0,6 to 0,8

The internal efficiency or acceptable efficiency is expressed as the ratio of the internal power to the thermal power supplied to the fuel.

$$ni = \frac{Pi}{Q},$$

$$Q = K * Hu$$

K = fuel consumption (quantity in unit of time), Hu = specific calorific value of fuel. The internal efficiency can also be determined by means of the thermal efficiency and the quality grade.

$$nth = \frac{PI}{Q},$$

$$ni = \frac{Pi}{Q},$$

$$ni = nth * ng$$

The mechanical efficiency includes all power losses due to friction, including the power necessary to transmit the motion of the auxiliary devices. It shall be calculated according to the relationship:

$$nth = \frac{\mathrm{P}\varepsilon}{Pi}$$

and its value is 80 %.

The net efficiency or active efficiency is the ratio of the net power to the thermal power supplied to the fuel:

$$n\varepsilon = \frac{\mathrm{P\varepsilon}}{\mathrm{KHu}}$$

This efficiency derived from the individual efficiencies is:

$$n\varepsilon = \frac{\mathrm{P}\varepsilon * PI * Pi}{Pi * PI * KHu} = nm * ng * nth = nm * ni \qquad (1.4.1)$$

The following optimum values are obtained for the net efficiency:

engine Otto engine nε = 0,25 to 0,30

diesel engine nε = 0,3 to 0,45

The higher optimum values can only be achieved for a specific operating condition. Otherwise, the efficiency is worse. In idling mode (net power is equal to zero), the net efficiency is also equal to zero.

The specific fuel consumption kε is that consumption related to power. As a reference power, the net power is usually used:

$$k\varepsilon = \frac{K}{\mathrm{P\varepsilon}}$$

For the above quantities, the units of measurement K, in g/h, Pε in kW are usually used, from which the unit of measurement g/k Wh is derived for specific fuel consumption. Using the formula used for the net efficiency, the following equation for specific fuel consumption is obtained:

$$n\varepsilon = \frac{\mathrm{P}\varepsilon}{KHu} = \frac{1}{k\varepsilon Hu} \rightarrow \kappa\varepsilon = \frac{1}{n\varepsilon Hu} \qquad (1.4.2)$$

Provided that the specific calorific value Hu for petrol and diesel is 42.000 kj/kg, the above relationship is further simplified:

$$\mathrm{k\varepsilon} = \frac{1}{\mathrm{n\varepsilon} * 42000 * \frac{\mathrm{kJ}}{\mathrm{kg}} * \frac{\mathrm{kWh}}{3600\mathrm{kJ}} * \frac{\mathrm{kg}}{1000\mathrm{g}}} = \frac{86}{\mathrm{n\varepsilon}} \left(\frac{\mathrm{g}}{\mathrm{kWh}}\right) \qquad (1.4.3)$$

By taking the mentioned useful efficiencies and the relationship (1.4.3), the following rounded values for specific fuel consumption are obtained:

Otto engine kε = 285 to 345 g/kWh.

diesel engine kε = 190 to 285 g/kWh.

1.5 Average Piston Pressure

The area of the illustrative diagram corresponds to the work delivered per engine cycle. If this area is divided into a rectangle of equivalent area (equal piston stroke), the height of the rectangle corresponds to the average piston pressure. With the aid of this pressure, the work output per cycle can be written for a cylinder as follows:

$$E = pi * Vh$$

pi = average piston pressure, average indicative piston pressure and Vh = piston volume. The average piston pressure can also be described as the specific piston work. According to the relationship pi= E/Vh, the average piston pressure is described as work per piston volume. The average effective piston pressure pε is a purely calculative quantity, just like the average indicative piston pressure pi. The equation of the net power is:

$$P\varepsilon = \rho\varepsilon * Vh * zni \qquad (1.5.1)$$

The effective power and the internal power differ from each other in terms of friction power and are Pε=Pi nm and pε=pi nm. The average effective piston pressure is obtained from equation (1.5.1):

$$p\varepsilon = \frac{\mathrm{P}\varepsilon}{\mathrm{Vh*zni}} \qquad (1.5.2)$$

For new engines, empirical values for the average active piston pressure are calculated by means of equation (1.5.2).

The calorific value of the mixture Hα is the amount of heat released by the combustion of 1 standard cubic meter of a mixture of petrol or diesel, gas and air.

The flow rate term π1 is the ratio of the actual mass m1 of fresh air mixture in the cylinder after filling to the theoretical filling mass mm*pα is the density of the fresh mixture at the pressure and temperature of the outside air and p1 is the density of the fresh mixture in the cylinder:

$$\pi 1 = \frac{p1}{p\alpha} = \frac{T\alpha * p1}{T1 * p\alpha}$$

Tα is the absolute temperature of the outside air. T1 is the absolute temperature of the mixture in the cylinder, pa is the absolute pressure of the outside air and p1 is the absolute pressure in the cylinder. The temperature T1 is higher than Ta because the fresh air mixture is heated on the hot walls as it enters the cylinder. The pressure p1 is less than pα because of losses during inflow. The assumption that the total piston volume is filled with the fresh mixture is verified for the four-stroke engine. In the two-stroke engine, the fresh mixture is partially mixed with the exhaust gases during scavenging, so that the total piston volume does not contain pure fresh mixture.

Empirical values:

Four-stroke engine π1 = 0,7 to 0,9.

Two-stroke engine with crankcase compressor and chamber compressor π1= 0,5 to 0,7.

To derive the equation for pressure pe, the relationship for net power is given:

$$Pe = Q * n\varepsilon = Q * nth * ng * nm$$

The amount of heat supplied per unit time is:

$$Q = Ha * m1 * p0 * zni$$

p0 is the normal density of fresh mixture with:

$$\mathrm{m1} = \pi 1 * mth$$

$$mth = pa * Vh$$

$$Q = \mathrm{H}\alpha * \pi 1 * \frac{pa}{\mathrm{p0}} * Vh * zni$$

Q is substituted into the power equation Pε:

$$P\varepsilon = nth * ng * nm * Ha * \pi 1 * \frac{p\alpha}{p0} * Vh * zni$$

For the useful power the relation is also known:

$$P\varepsilon = p\varepsilon * Vh * zni$$

The two relations for the power Pε are equalized, so that the required equation for the average active piston pressure is obtained:

$$p\varepsilon = nth * ng * nm * \pi 1 * H\alpha * \frac{p\alpha}{p0} \qquad (1.5.3)$$

Since relation (1.5.3) is an equation of quantities, pε has the same units as the calorific power of the mixture.

CHAPTER 2

2 Combustion

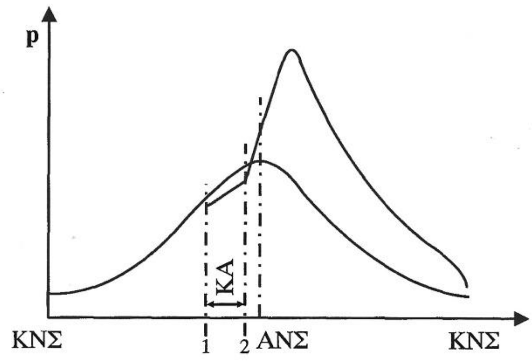
2.1 Ignition and Ignition Systems

The combustion of a fuel-air mixture takes place through ignition. To achieve ignition, the mixture must be flammable, its composition must be within the ignition limits. The ignition limits are expressed in terms of the percentage (%) by volume of fuel (gaseous state) in the mixture and depend on the type of fuel and temperature. In addition, at least at one position in the mixture the ignition temperature must be reached or exceeded. Ignition temperature is the lowest temperature after which ignition occurs and is determined by the type of fuel and the density of the combustion air. It decreases with increasing pressure. To prevent the flame from being extinguished, the heat of combustion drawn must maintain at least the minimum ignition temperature. Fuel efficiency limits for petrol are 1,4 for a poor blend, 7,0 for a rich blend and ignition temperatures of 480 to 550 degrees Celsius.

Ignition can be achieved in two ways, by external ignition or self-ignition. External ignition is used in the Otto engine and self-ignition in the diesel engine. In external ignition a foreign energy source is required to provide the necessary heat and temperature, whereas in the Diesel engine the ignition temperature is achieved through the high compression of the fresh mixture.

In the external ignition mode, high voltage ignition was developed by Bosch, which is still used in engines. In the diesel engine, the temperature of the charge in the cylinder is raised by high compression to values higher than the ignition temperature of the oil. When the piston is just before top dead center , oil is injected into the compressed and high temperature air in the cylinder and the mixture ignites. When the engine is very cold, then the ignition temperature can be reached by compression. In this case, starting igniters offer help. Before starting the engine, the ignitor circuit is connected for a few seconds to a minute or so and the ignitor's glowing spiral (rod) heats the combustion chamber.

An illustrative diagram shows that combustion does not take place now of ignition or injection, but at a short time afterwards (Figure (2.1.1)).



(Figure (2.1.1)) Ignition delay in a diesel engine

The beginning of combustion is distinguished by the sharp increase in pressure that clearly deviates from the multimodal compression curve. The time interval between the ignition or injection timing and the start of combustion is called the ignition delay. The time duration is approximately 1/1000th of a second. During the time of the ignition delay, the fuel is prepared for combustion, vaporized and chemical pre-reactions take place. The duration of the ignition delay depends on the type of fuel, temperature, and pressure. Oil is composed of large molecules of hydrocarbon compounds that are easily dispersed. Its ignition delay is short. Petrol, especially Super, consists of hydrocarbon compounds with a long ignition delay. With increasing pressure and temperature, the ignition delay becomes shorter. In the diesel engine a shorter ignition delay is desirable. The oil must be burnt immediately after the passage into the cylinder, to influence the combustion pressure through the incoming quantity during injection. In the case of a long ignition delay, the entire quantity of injected oil would be explosively burned by a high-pressure build-up at high pressure. Also, the fuel used in the Otto engine must have a long ignition delay to avoid pre-ignition accompanied by shock combustion.

Ignition systems are subdivided into battery ignition systems and magneto ignition systems. Battery-operated ignition systems, because they provide the strongest spark at start-up and at low revs (which is why the battery is essential for starting the engine), are mostly used in Otto engines. Magnetic ignition installations are used in small cheap engines, and also where battery maintenance is not ensured. Conventional coil ignition has now been replaced by electronic ignition installations with a high voltage capacitor. Electronic components have replaced the mechanical switch of the common manifold. Because electronic ignition installations are not subject to wear and tear, they do not require maintenance and the ignition timing once set does not need to be readjusted. The electronic ignition systems shall be operated in such a way that the ignition voltage is reduced to a minimum with increasing speed and a large amount of energy is available for ignition of the mixture. By adjusting the ignition characteristic field, the engine can be operated over the entire power-rpm range at the optimum ignition timing points.

The conventional coil ignition consists of the battery (1), ignition circuit breaker (3), ignition coil multiplier(4), contact switch (plates) (6), capacitor (5), distributor (7), and igniters or spark plugs (8) (figure (2.1.2)).

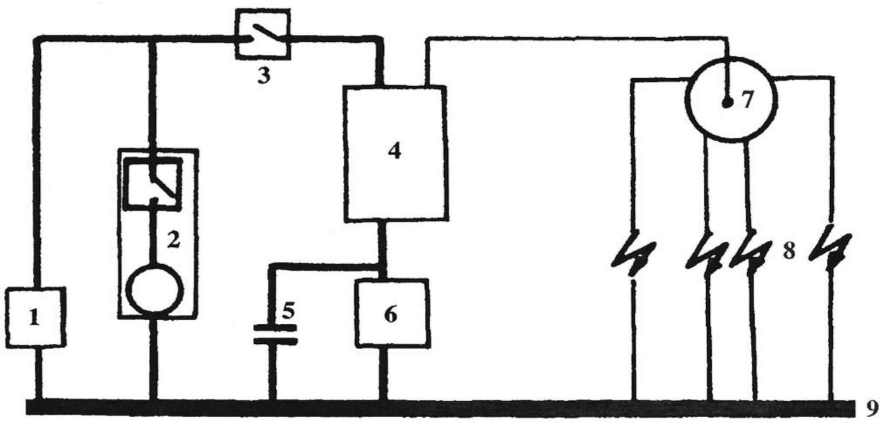


Figure (2.1.2) Coil ignition.

In parallel with the battery is connected the mechanism for lighting (2), which by means of a regulator switch, is then connected to the mains only when its voltage with increasing speed exceeds the battery voltage. The ignition coil is a transformer with a detailed secondary winding in relation to the primary winding. The distributor connects the secondary winding to the igniter with a rotating contact arm (roller) at the exact moment of sparking. When the ignition switch is activated, the primary current flows with the contact switch closed. The consequence of this is the creation of a magnetic field in the multiplier. At the moment of ignition, the camshaft raises the magnetic field.

This results in the creation of voltages in both windings. The primary voltage generated amounts to approximately 350 V and the secondary voltage, depending on the resistance ratio, to approximately 70 to 250 kV. This voltage results in the creation of a spark in the igniter. After ignition, the contact switch (platinum) is closed again, and the process is repeated. The duration of the contact switch closure is expressed in degrees of the distributor-camshaft angle and is called the closure angle. The angle of closure depends on the geometry of the cam and the distance of the contact switch (plate gap). The closing time decreases with increasing speed. Since the primary current reaches its maximum value after a fixed time, the final value of the current is influenced by the closing time and therefore the rpm of the motor. Less current has the physical consequence of a weaker magnetic field, which provides a lower voltage. Hence the secondary voltage decreases as the speed increases. Also, at low and high speeds, the current is not interrupted frictionlessly. At low speeds power sparks occur at the contacts of the boards, which slow down the magnetic field interruption and thus reduce the ignition voltage. The large voltage drop at high speeds occurs due to the sharp bumps at the contacts. The maximum number of sparks is therefore limited and is limited to about 18000 sparks/minute.

Coil ignition has the disadvantages of generating a strong drop in ignition voltage when speed is increased and the contact switch connecting the full and primary currents is subject to relatively high wear. The coil-transistor ignition is so designed that the voltage drop with increasing speed remains small. A transistor is an elementary semiconductor. In the ignition installation, it is placed as an electronic switch (12) in the primary circuit (figure (2.1.3)), and connects and disconnects, instead of the mechanical switch (4), the primary current. The transistor requires a small amount of current for its operation for its regulation. When the regulating current is flowing, then the transistor is open for the primary current, and if the regulating current is interrupted, then the transistor stops the primary current. Initially the buffer current closed the circuit or disconnected by a mechanical switch(pulses), just like in conventional coil ignition, and we had the adjustable contact coil-transistor ignition. Today the buffer current is generated by electric pacemakers, and we have the adjustable (contactless), full electronic coil-transistor ignition.

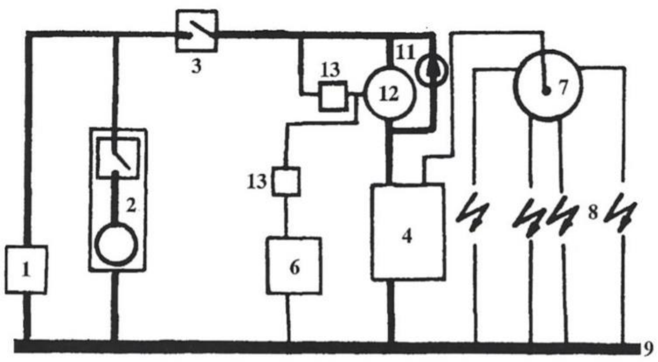


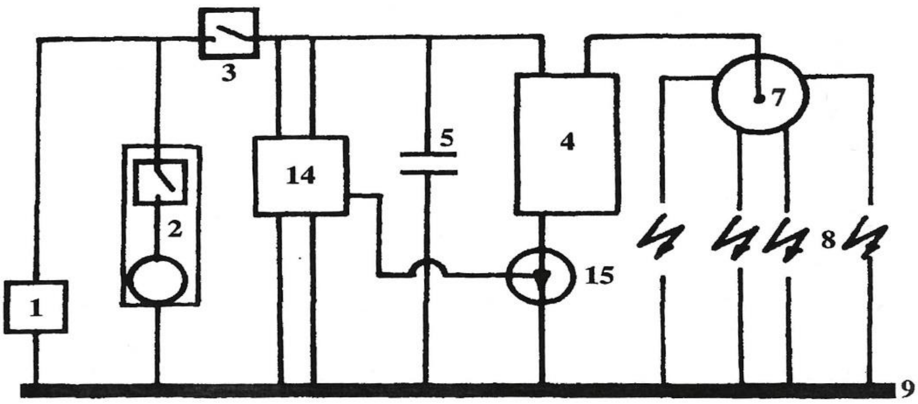
Figure (2.1.3) Adjustable ignition with coil transistor.

Because there is no mechanical contact in full electronic ignition with coil-transistor, there is no mechanical wear and tear, so no maintenance is required. Also, the ignition timing remains stable even at high rpm. In coil-transistor ignition a multiplier (4) with minimum inductance is used in the primary circuit with the primary current (about 9A) being about twice as high as in coil ignition. Due to the low primary inductance, the primary current increases faster and the final current value is higher. For this reason, in coil-transistor ignition, when the speed increases, the ignition voltage does not decrease as much as in conventional coil ignition, while more ignition energy is available with a higher primary current. To protect the transistor, a Zener diode (11) is connected in parallel. It is an elementary semiconductor which, when the voltage limit is exceeded, immediately shuts down the transistor. To generate the buffer current of the transistor, which closes the primary current, the inductive Hall inductor is usually used in full electronic ignition. The inductive donor is mounted in the ignition distributor and can also be mounted directly on the crankshaft. The Hall effector is manufactured inside the ignition distributor.

The ignition voltage and the energy available for ignition can be improved by electronically adjusting the closing angle. Without the adjustment , the closing angle has a fixed value, and the final value of the primary current is further reduced by increasing the rpm or decreasing the battery voltage. This can be avoided by electronically adjusting the closing angle to keep the primary current value constant. In this way , the voltage and ignition energy remain unchanged, with high values. Of course, the increase of the closing angle is limited by the duration of the ignition spark. A too large closing angle would reduce the opening of the angle and at the same time the duration of the ignition spark too much, so that a safe ignition system is no longer provided.

The advantages of coil-transistor ignition over coil ignition can be summarized in that it is maintenance free, has a constant ignition timing, a smaller drop in ignition voltage with increasing rpm, higher ignition energy and a maximum number of sparks, around 30,000 sparks/minute.

In the multiplier ignition mechanism, the ignition energy is stored in the magnetic field. The storage of ignition energy with a high-voltage capacitor is the capacitor with its electric field. The basic reason for the construction of high-voltage capacitor ignition is to increase the ignition voltage by an order of ten or so higher than the value of coil or transistor-coil ignition (about 3000 V/μs in capacitor ignition and about 4000 V/μs in coil or transistor-coil ignition). With a sharp increase in voltage, the energy losses due to the connections to the igniters remain small and have no effect on the ignition voltage. Ignition with a high voltage capacitor is particularly preferred where a lot of dirt is deposited on the igniter spike due to oil soot and others. The schematic of the construction of high voltage capacitor ignition is shown in drawing (2.1.4) .



Drawing (2.1.4) High voltage ignition with capacitor.

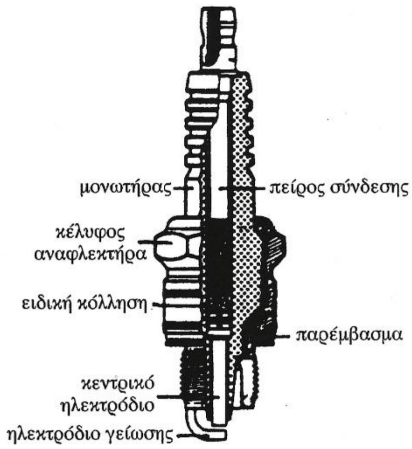
In the primary circuit of the ignition installation there is the accumulator capacitor (5); from the charging installation the capacitor is charged before each ignition with about 400 V. At the moment of ignition, the electronic switch (15) of the primary circuit is closed and the capacitor is discharged through the primary winding of the multiplier (4). In this way the ignition voltage is developed in the secondary winding, which can reach up to 30 kV. In high-voltage capacitor ignition, the burning time of the ignition spark is shorter than in coil ignition installations.

The spark duration has no effect on the ignition of a uniform mixture with the correct composition. However, when the fuel and air are unevenly distributed, or when the mixture is too rich or too poor, a short duration spark may result in combustion failure. The drop in ignition voltage with increasing rpm is less in high-voltage capacitor ignition, as opposed to conventional coil ignition.

It is generally concluded that the conventional coil ignition is replaced by the full electronic ignition with coil-transistor ignition. High-voltage capacitor ignition is used in special cases (where the mixture is not fully ignited due to residual ignition residues in the igniters). Full electronic ignition with coil-transistor ignition is maintenance-free, and the adjustable ignition timing remains constant without resetting. The fully electronic ignition with coil-transistor provides the possibility of adjusting the closing angle, through which the ignition voltage remains uniformly high regardless of engine speed and ignition power supply. In this way, poor gasoline-air mixtures can also be ignited well.

A high-voltage spark is generated between the two electrodes of the igniter now of ignition, which ignites the fuel-air mixture. The igniter consists of the following parts: (Figure (2.1.5)): connection pin and central electrode, insulator, igniter shell and earth electrode.

The cable coming from the distributor is connected to the steel connecting pin. The central electrode and the connecting pin are connected by an electrically conductive special bonding and are airtightly surrounded by the insulator. The central electrode is affected by the high combustion temperature and is also exposed to the strong corrosive atmosphere of the gases and combustion residues. To keep wear and tear to a minimum, it is made of nickel alloy with additions of chromium, manganese, and silicon.



Drawing (2.1.5) Igniter

The insulator is recommended to be made of aluminum oxide with glass additives. It shall have high resistance against ignition voltage, as well as high thermal and mechanical strength. In the operation of the engine under full load, the temperature developed at the bottom of the insulator is about 850 degrees Celsius and at the head of the insulator about 200 degrees Celsius. In addition to this, the lower part of the insulator is exposed to sudden temperature changes in the combustion chamber. The heat dissipation capacity of the insulator must be good to reduce the high temperatures of the lower part. Also, external coils increase the resistance to spark transmission from the connecting pin to the igniter shell.

The insulator is sealed with drip rings and wraps in the steel shell. The grounding electrode consisting of the same material as the main electrode is welded to the shell. To fit the motor, the shell is threaded and gasketed. The pattern and arrangement of the earth electrode is adapted according to its intended use (for use in a two-stroke, four-stroke or racing engine). The electrode gap is 0,7 to 0,9 mm for battery ignition systems and 0,4 mm for magneto ignition systems.

In addition to the standard air gap spark igniters, slip spark igniters are available for special applications. In these igniters the spark slides across an insulating layer from one electrode to the other. The way they are constructed gives them the advantage that the combustion residues do not affect the course of the spark. Their disadvantage is that the mixture does not diffuse as well in the spark area. To overcome the high voltage requirement, the slip spark igniter is usually equipped with an adjustable electrode. In a cold igniter a spark is generated between the main electrode and the adjustable electrode. As the temperature of the insulator increases, the voltage requirement for the slip igniter path decreases and the spark jumps there. Due to the change in the spark path, the wear of the electrode is less, and the service life of the igniter is long. In addition to this, this specially designed igniter has the advantage, due to the long spark path, of better ignition of rich and poor fuel-air mixtures.

In the conventional ignitor , the bottom of the ignitor must have a high temperature to burn the residual fuel-oil, otherwise leakage paths for the current are created and the operation of the ignitor is interrupted. The lower temperature limit for the combustion of the fuel-air residues in the lower part of the insulator is about 500 degrees Celsius. Of course, lead deposits are not eliminated, which could also lead to an ignition failure with a superheated igniter. When maintaining the igniter, not only the diaphragm should be adjusted by bending the electrode, but also the insulator should be cleaned with a special cleaning device. The upper operating temperature of the lower part of the insulator shall not exceed 920 degrees Celsius. Otherwise, ignitions due to glowing occur. Glow ignition occurs irregularly during ramming due to glowing bottom of the insulator and the engine power drops and the igniter may be damaged due to overheating. Of course, ignitions due to glowing can also be created in the combustion chamber by glowing combustion residues. The igniter must have a heating capacity, according to the installation engine, so that the temperature of the bottom of the insulator is in the desired range of 500 to 900 degrees Celsius. The typical heating power rating on a newly constructed engine shall be calculated. An igniter with a low heating power characteristic number has a small heat absorption area and is indicated for high stressed engines. In low stress engines, igniters with a high heating value characteristic are used, which means a large heat absorption surface.

2.2 Timing and Knocking (experiments) in Internal Combustion Engines

The ignition timing is that at which an ignition spark is generated in the igniter. It is given in degrees of crankshaft angle with reference to the position of the top dead center of the piston. The ignition timing is not a fixed value but is adjusted to the engine operation in order to achieve perfect combustion of the fuel mixture. In conventional ignition systems with manifolds, the ignition timing is shifted mechanically (Figure (2.2.1)). With increasing speeds, the ignition timing is shifted by the centrifugal governor to an early position to compensate for the increasing ignition retardation.

Also, a pressure-operated diaphragm through the intake manifold changes the ignition timing according to the engine load. When the engine is operating in the low-mixture range (λ = 1,1), ignition must be affected earlier than in the case of full power operation (λ = 0,9). In the case of idling and when the vehicle is moving the engine, ignition is affected later. Under these operating conditions the throttle valve in the suction duct should be closed. However, this would result in a poor mixture and intermittent combustion. For this reason, the throttling valve remains open. So, in order not only to increase the rpm, but the ignition is also shifted accordingly.

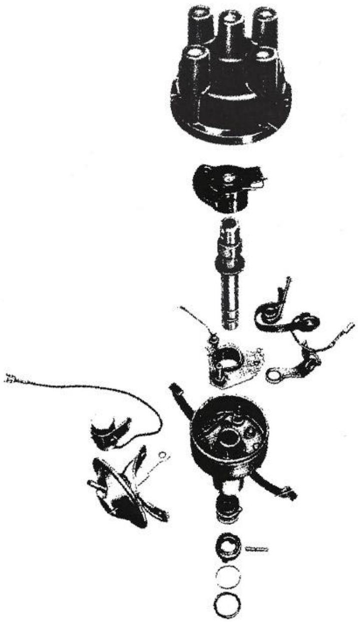


Figure (2.2.1) Mechanical shift of the ignition timing (ignition distributor).

With the electronic ignition shift, the ignition is even better adapted to the desired engine operating range. The rev- and power-dependent ignition characteristic field of the electronic ignition offset is more precisely accurate compared to that of the mechanical ignition timing offset. It consists of 4000 values for the ignition timing, which are determined in the running test and then stored electronically. To set the optimum ignition timing for the engine operating conditions, a microcomputer gives the corresponding value from the characteristic field. This value in combination with other data, such as engine temperature, air temperature, throttle valve position, is then corrected by the microcomputer and transmitted to a fully electronic ignition system.

During combustion, harmful substances are present in the engine and are emitted together with the gases into the environment. To reduce the environmental impact of the means of transport, emission limits for pollutants have been legislated. A properly designed and regulated ignition system contributes significantly to the reduction of polluting emissions. The following harmful substances can be detected in exhaust gases: carbon monoxide, hydrocarbons, nitrogen oxides, lead compounds and soot. The quantity of hydrocarbons in the exhaust gas is particularly affected in poor mixtures by the duration of the spark. An ignition system with a long spark duration result in the emission of exhaust gases with fewer hydrocarbons. Conventional coil ignition and coil-transistor ignition have an advantage over those of the high-voltage capacitor because of this. It should be mentioned that high-voltage capacitor ignition is less susceptible to impure igniters and thus contributes to emission reduction. The position of the ignition timing plays a role in the emission of hydrocarbons and nitrogen oxides. Late ignition reduces the emission of these harmful substances. Poor combustion engines burn very poor fuel-air mixtures ($\lambda>1,2$). Fuel consumption is low and smaller quantities of harmful substances are emitted. However, to ignite these mixtures in the ignition plant, there are increased requirements, such as higher ignition voltage, longer ignition duration and spark energy and early adjustment of the ignition timing.

During the rapid percussive combustion of the fuel in the engine cylinder, a diffuse rattling sound can be heard in the Ottos and a harsh knocking sound in the Diesels. This knocking noise occurs due to strong pressure waves in the cylinder, which impinge on the cylinder head and piston. During combustion in an engine in which such knocks occur, the piston is subjected to considerable thermal and mechanical stress. Constant knocking leads to piston corrosion. At high temperatures the piston expands, and a thin layer of lubricant is compressed on the cylinder walls. At some locations on the walls, due to high friction, temperatures are so high that fusion welds occur. Because of the movement of the piston, cracks appear at the locations of the welds and the roughness between the piston and the cylinder walls increases to such an extent that the piston eventually fuses and stalls. This results in major engine damage. Also, in the case where no piston corrosion occurs (in the case of a stroke engine), the result is a very high load on the transmission, a high overheating of all parts in contact with the combustion gases of the engine and a drop in engine power. For this reason, an engine must run without knocks.

In normal combustion in the Otto engine, the fuel-air mixture is ignited in the igniter and the flame spreads in the form of a spherical wave in the combustion chamber, with an average speed of about 20 m/sec. Also, in combustion in the presence of shocks the mixture is ignited by the igniter, then the temperature and pressure of the burning gas are increased, and the pressure waves propagate through the combustion chamber. In the non-burning mixture, the temperature and pressure also increase. At some locations when the auto-ignition temperature is exceeded, ignition sources are formed, resulting in shock combustion of the remaining mixture and the occurrence of strong pressure waves, which when they hit the walls, cause a strong shock noise.

The measures to be taken to avoid shocks have to do with whether the engine knocks during movement (running); whether they should be avoided in the design of the engine and whether they should be avoided in the design of the engine and whether fuels with a high resistance to shocks should be prepared.

When the vehicle is on a course and the knock occurs due to acceleration or high speed. In both cases, the engine is subjected to high stress. Acceleration shocks occur when accelerating the vehicle from low rpm using a full fuel mixture flow. Here it would be helpful to select the next lower speed, with the result that for the same power output of the engine the engine speed would increase, and the torque would decrease. The filling of the engine becomes smaller because the throttle valve in the suction pipe is lying a little more, the compression becomes smaller, and the knock disappears. High-speed knocks occur in operation under high rpm. In this case, no auxiliary measures can be taken, and rarely does this phenomenon lead to piston destruction. When the rpm is reduced, the knocking stops. When knocking occurs in normal-benzine operation , then the use of benzine-super eliminates the problem. Also, the tendency of an engine to exhibit knock is reduced when the ignition timing is shifted to an ignition delay position. With the help of the ignition delay the pressure in the cylinder remains lower and the tendency of the fuel to self-ignite is also lower. It goes without saying that in this way the engine power is reduced, and fuel consumption is increased.

If measures must be taken to avoid knocks when designing the engine, the following options are available to the manufacturer: choice of compression ratio, position of the igniters and shape of the combustion chamber.

The compression ratio is selected in accordance with the petrol available on the market for anti-current engine operation, normal-compression ratio petrol up to ε=9 and super petrol from 8,5 to 10. When selecting the compression ratio, it should be noted that a high compression ratio e increases engine power and reduces fuel consumption.

The tendency to cause knock in an engine is reduced when the flame moves from a hot mixture to a colder mixture. The hottest position in the combustion chamber is the exhaust valve and the igniter should be located close to it. The last part of the fuel mixture should be kept at a low temperature by properly cooled combustion chamber walls to prevent premature spontaneous combustion.

Also, the shape of the combustion chamber is affected by the hits. A solid combustion chamber reduces impacts more than one with many protruding surfaces and flanges. And air turbulence creates a uniform fuel mixture and uniform temperature distribution. The flame travels through the combustion chamber faster and there are no combustion shock reactions. By appropriate configuration of the intake ducts or corresponding configuration of the combustion chamber and pistons, the mixture is swirled. With good cooling, the mixture remains cold and less flammable. The water-cooled engine is preferable here to the air-cooled engine. By using aluminium alloys instead of cast iron, temperatures at the cylinder head remain lower(three times the thermal conductivity of aluminium alloys compared to cast iron).

The measures taken to avoid shocks by producing high-strength fuels have to do with the following. The fuels are obtained by fractional distillation of the oil recommended by a variety of hydrocarbon compounds, which exhibit a completely different impact resistance. The distilled oil is subjected to chemical processes designed to enrich it with impact-resistant hydrocarbons. Thus, in the boiling range from 40 to 200 degrees Celsius (maximum boiling point 215 degrees Celsius), normal and super petrol are obtained with approximately the same heating capacity. Super petrol has more shock-resistant hydrocarbons than normal petrol. Super petrol has a density of 0.74 g/cm^3, both fuels are available with or without added lead. While unleaded normal petrol has the same octane number as leaded petrol (about 92 octanes), the octane number of super is about three points lower than that of leaded super (octane number 98). Unleaded petrol is necessary in catalytic engines. In the case of leaded petrol, the maximum permissible value is 0,15 g/l, this addition serves to increase the resistance to shocks. As anti-corrosive agents, tetraethyl lead ($Pb(CH_3)_4$) and tetraethyl lead ($Pb(C_2H_5)_4$) are used. Both lead compounds are poisonous. Their effect is based on the fact that due to the high temperature; they decompose before combustion and the resulting lead dust prevents premature spontaneous combustion. To prevent the formation of lead oxides during combustion, which would cause an increase in cylinder wear, chlorine or bromine compounds are added to the benzines. The lead is burned to produce lead bromide or chloride. These are highly poisonous lead compounds at about 800 degrees Celsius, are in a gaseous state and are emitted with the exhaust gases from the engine. In addition, they are included among the harmful substances in exhaust gases and contribute to environmental pollution. State laws have reduced the limits of lead additives in petrol and in the future, petrol with lead additives will be completely replaced by unleaded petrol. Adding alcohol to gasoline such as methanol increases shock resistance. In this case, however, with an adsorbent of more than 15%, the mixture formation facilities must be harmonized, especially in the petrol-alcohol mixture.

The impact resistance of petrol is reflected in its octane number. The octane number indicates that the petrol is as shock resistant as a defined comparative blend of isooctane and normal heptane. However, because isooctane has a specified octane number of 100 and normal heptane has a specified octane number of 0, for example, a petrol with an octane number of 80 means that the petrol is as shock resistant as a comparative mixture of 80 % by volume of isooctane and 20 % by volume of normal heptane. The impact resistance of gasoline increases with increasing octane number.

The determination of the octane number is carried out in special control engines. Permitted control engines are the CFR control engine (Cooperative Fuel Research Committee of the American Society of Automotive Engineers) and the BASF control engine (Badische Aniline- und Soda-Fabrik). The control engine is a single-cylinder four-stroke engine, where the compression ratio must be maintained during operation between 4 and 11.

In the diesel engine, the fresh air is compressed so much that its temperature reaches higher points than the ignition temperature of the oil. Just before the piston reaches top dead center the fuel is sprayed. The amount of oil injected into the aerator during the delayed ignition is burned percussively. If this quantity is large, strong pressure waves occur which produce a shock noise. Usually, these shocks are particularly loud in cold engines running at idle (idling), or under partial load. This is due to the long ignition delay, which becomes much shorter with increasing pressure and temperature. The knocks in idle mode (idling) are not dangerous for the engine and disappear with increasing load. In an engine with direct injection of oil into the combustion chamber air, knocks are avoided if the injected fuel quantity is kept low during the ignition delay. The main quantity is supplied immediately after the start of combustion. In this case, the disadvantage of the presence of soot cannot be avoided. Soot occurs when there is not enough time for the fuel to evaporate and mix with the air before combustion. Particularly when the pressure and temperature are high and only a small amount of air is available for combustion, decomposition reactions take place which lead to the formation of soot. Since the soot is not fully combusted, it leaves the engine with the exhaust gases.

Shock combustion of the fuel can also be suppressed by separation of the combustion chamber. The oil is sprayed into a space separate from the cylinder. Only a portion of the fuel is burned there due to lack of air. The pre-combustion in the separate space increases the temperature and pressure. The remaining unburned fuel passes through the passage into the cylinder space at high speed and the combustion of the remaining mixture takes place there. Due to the time prolongation of combustion, even in fuels with a long ignition delay, no knocks occur. However, this advantage is offset by the fact that fuel consumption increases. In addition to the above two methods of mixture formation, in which the fuel is sprayed into the air, there is another different method in which the fuel is sprayed into the air, there is another different method developed by the MAN company. The fuel is sprayed as a thin film onto the walls of the combustion chamber (piston). In this method of mixture formation, no flicks occur because the fuel burns to the extent that it evaporates from the walls and mixes with the swirling air. Engines that operate according to this method of mixture formation are called multi-fuel engines, because they can burn fuels ranging from lubricating oil, diesel and even petrol. There are other ways of avoiding flicks. Diesel oil is obtained by fractional distillation of crude oil in the boiling range of 200 to 360 degrees Celsius and contains many paraffinic hydrocarbons of high flammability. The addition of ignition accelerators further increases the ignition propensity of the oil. The effect is because by introducing them into the hot air they burn directly and by increasing the temperature the ignition delay of the oil is reduced. It is sufficient to add ignition accelerators to diesel oil in a proportion of 0,1 to 1 % by volume.

The ignition tendency (flammability) of diesel oil is expressed in terms of the number of cetane numbers. The cetane number expresses the fact that diesel oil has the same tendency to ignite as a defined reference blend of ketane and methyl naphthalene. The recommended flammability of the mixture is ketane and was given a value of 100 and the non-flammable methylnaphthalene was given a value of 0. Thus, a cetane number of 55 means that the diesel oil has the same flammability as a comparative blend containing 55% by volume of ketane and 45% methylnaphthalene. The flammability increases with increasing cetane number.

The determination of the cetane number is obtained by the same procedure as the octane number in the case of dedicated control machines. The BASF, the CFR control motor is used. Both engines are single-cylinder, four-stroke diesel engines with compression-regulator. In the BASF control engine, the compression is adjusted by throttling the intake air and in the other engine, by varying the compression ratio. The cetane numbers of today's diesel are between 50-55.

CHAPTER 3

3 Fuel Mixture Formation and Emissions

3.1 Fuel Mixture Formation in the Otto Engine

In the process of fuel mixture formation, an ignitable homogeneous mixture capable of complete combustion is necessary. Perfect combustion, in which the hydrocarbons react with oxygen to give carbon dioxide and water, is achieved by two main factors: the complete conversion of all the chemical energy of the fuel into heat and the reduction of exhaust gas emissions (so that carbon monoxide, hydrocarbons and soot are not present in the exhaust gas). Perfect combustion is only achieved when the air ratio is λ=>1. Ideally, each elementary quantity of fuel must be supplied with a reciprocal quantity of oxygen. For this reason, the easiest and fastest way is to mix oxygen fuel in the gaseous state. This justifies the superiority of gaseous fuel combustion engines in terms of the degree of complete combustion and emission reduction.

Also in liquid fuel engines ,in order to achieve complete combustion, the fuel must be converted to a gaseous state. Because diesel oil has a high boiling temperature, the formation of a fuel mixture presents great difficulties. The chain molecules of the hydrocarbons in diesel fuel tend to break down under high temperatures and pressures. The result of these decompositions is the formation of soot, which is not fully combusted and gives the familiar dull color to diesel exhaust gases.

Most Otto engines use petrol as fuel. Nowadays, gas engines are becoming interesting again for two reasons: a) The discovery of new deposits of gaseous hydrocarbons makes it a cheap fuel for gas engines, b) In gas engines it is much easier to meet emission standards.

The formation of the fuel mixture starts from the ventilator or injection facility and tends to the cylinder during compression.

The purpose of the ventilator or injection system is to regulate the amount of air so that a homogeneous fuel-air mixture is achieved. The suction lines between the vaporizer and the intake valves must be configured in such a way that all cylinders are filled with an equal amount of fuel mixture. Injection installations allow a freer design of the suction lines because the fuel is either injected just before the valves or directly into the cylinder. The fuel mixture formation is further improved by the appropriate configuration of the intake duct and combustion chamber.

The ventilator is incorrectly so called because it is generally through it that the fuel is nebulized. In the past, during the development of the first engines, there were real carburetors in which the air was mixed with gaseous fuel. The drawing (3.1.1) shows the following components of the ventilator: Suction duct, starting valve, air hopper, regulating valve, petrol tank, float, needle valve, central nozzle, and injection pipe. The air sucked from the engine flows into the suction duct. In the air hopper, which consists of the nozzle, the cylinder section and the diffuser, fuel is added to the air from the injection pipe. The throttle valve regulates the amount of mixture entering the engine and at the same time the engine torque. The fuel enters the petrol tank through the needle valve. The bow adjusts the fuel level so that fuel does not flow out of the injection pipe when the engine is not running. The central nozzle influences the composition of the mixture. For a good dispersion of the fuel a high air velocity at the air hopper nozzle is necessary. At the same time, negative pressure is developed to expel the fuel from the injection pipe. By venting the petrol tank, the same pressure prevails as at the beginning of the suction pipe. When starting a cold engine, the starting valve remains closed. The engine sucks in less air, but much more petrol. This rich mixture is capable of ignition at low temperatures. During normal engine operation, the starting valve gradually opens.

As regards the relationship between the diameter d1 of the central nozzle and the diameter d2 of the air funnel, the basic relationship applies:

$$d1 = d2 * \left(\frac{0.050}{0.053}\right) \tag{3.1.1}$$

Exhausters are classified according to the direction of flow of the mixture, the type of construction of the mixture control device, the regulation of the flow of the lubricant and the number and type of construction of the suction ducts.

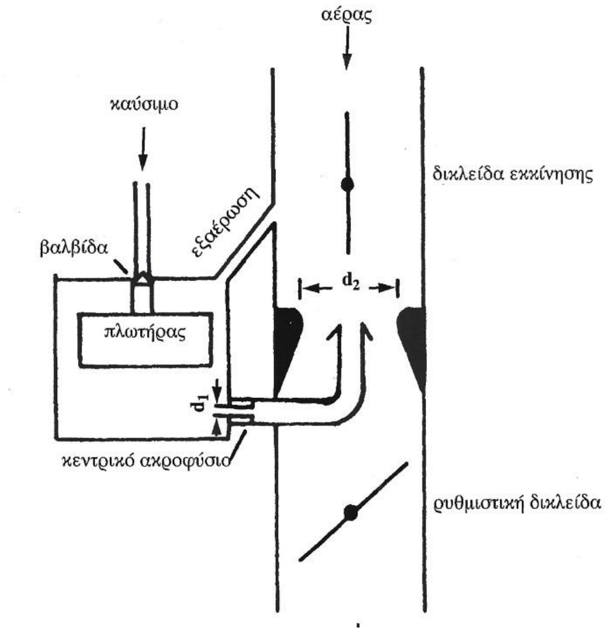


Figure (3.1.1) Simple ventilator.

Based on the classification of flow direction, three types of aerators are distinguished: downdraft, flat and updraft. Currently, only downdraft and updraft diffusers are manufactured. The flat-flow breather is suitable for low-altitude engines. By using multiple level exhausters, good cylinder filling is achieved because the mixture stream follows minimal bypass paths. Most engines use downdraft breathers. In these the mixture stream moves in the direction of gravity. This results in a good filling of the cylinder and the fuel tension, which escapes in the form of small droplets, is reduced to a minimum. The quantity of the mixture is regulated by regulating valves or steam gates. Cylindrical or flat steam gates are used. The cylindrical damper is widely used in motorcycle exhausts, although it has recently made its appearance in cars as well; when the position of the damper is varied, the air velocity remains almost constant. For this reason, fuel vaporization is uniformly good throughout the entire range of engine revs and load. The torque shows a desirable behavior and the ventilator operates in a vacuum up to the maximum power output without auxiliary systems. Flat latches are usually used in high-speed engines, mostly with petrol injection. With one latch, several intake manifolds can be evenly adjusted and under full load conditions the inflow of the mixture is not obstructed. The passage of the fuel through most ventilators is usually regulated by two floats. They exist in lawn mowers and other devices where we need to have operating autonomy from a positional point of view.

Depending on the number and type of construction of the suction ducts, ventilators are classified into three categories: The single, the stepped and the multiple ventilators. The cheapest in terms of construction is the single damper ventilator, which has only one suction duct. However, it has the disadvantage that under full fuel supply and high rpm, it cannot achieve good cylinder filling and consequently the piston power remains low. Better filling, over the entire operating range, is obtained with the stepped exhaust, which has two parallel suction ducts with separate throttle valves. In the low-demand-mix operating range, fuel is introduced only through one duct, while the other remains closed. At high load conditions the second one is also opened. The inflowing mixture is distributed in the two suction ducts and the cylinder is well filled under full load and high speed (maximum power output). The multiple exhaust consists of two or three single exhausts, assembled in one body and using the same float chamber. The suction ducts do not open one after the other, but at the same time because the dampers are connected in parallel. In the construction of, for example, a four-cylinder engine with two double exhausts, each cylinder must be filled by a suction duct. The advantage of this construction is that all cylinders are filled with the same amount of fuel and the mixture flow is minimally branched. For this reason, multiple exhausts are particularly suitable for high performance engines.

The following requirements are obtained from a single carburetor: fast and safe starting, low idling, more economical partial power consumption, faster response under acceleration conditions and a richer mixture for higher performance in peak power mode. The fuel flow rates, and mixture formations are adjusted so that at no point in operation are harmful substances (pollutants) in the exhaust gases exceeded. At low fuel consumptions the ratio shall be λ=1.1, as shown in figure (3.1.2).

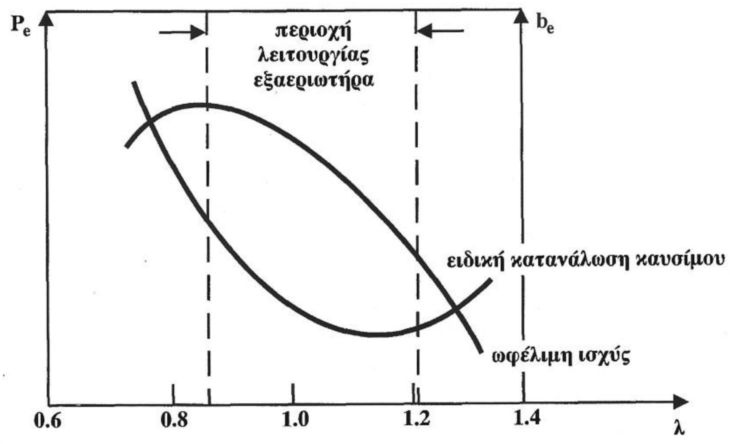


Figure (3.1.2) Dependence of specific fuel consumption and net power on the air ratio with constant throttle valve position and number of revolutions.

To achieve maximum power, an air ratio λ=0.9 is necessary, because this value corresponds to the maximum speed of the flame in the combustion chamber. It goes without saying that a ventilator, to meet all the above, that it will be a complex construction. For starting the engine, we must distinguish between cold and hot starting. In a cold engine, to prepare an ignitable mixture, the initial mixture must be rich in gasoline, which evaporates at a lower temperature. To make the mixture rich, two methods are used, that of the automatic starting throttle and the use of a starting ventilator. In both cases, the mixture is enriched in fuel by keeping the throttle in the closed position. For the convenience of the driver and for safe, fast, cold starting and with the correct fuel supply in warm operation, almost all carburetors today are equipped with an automatic starter.

All these additional requirements and additions make the ventilator more expensive to build but reduce emissions to the environment. High demands on mixture formation, e.g., low fuel consumption, reduction of harmful substances and good engine operation under all operating conditions, led to the development of the electronic mixture formation system "Ecotronic", (Figure (3.1.3.)). It consists of a ventilator, microprocessor, and sensors. Here, the throttle is not only dependent on the accelerator pedal but also on an electro-pneumatic mechanism. This mechanism acts on the initial valve as a pre-regulatory valve. Sensors receive data such as the number of revolutions, the engine temperature, the temperature of the suction pipe walls, the position of the throttle, the speed at the position of the throttle and supply them to the microprocessor. He processes and controls the electro-pneumatic mechanism. In figure (3..1.3.) the operation of the Ecotronic - ventilator is given.

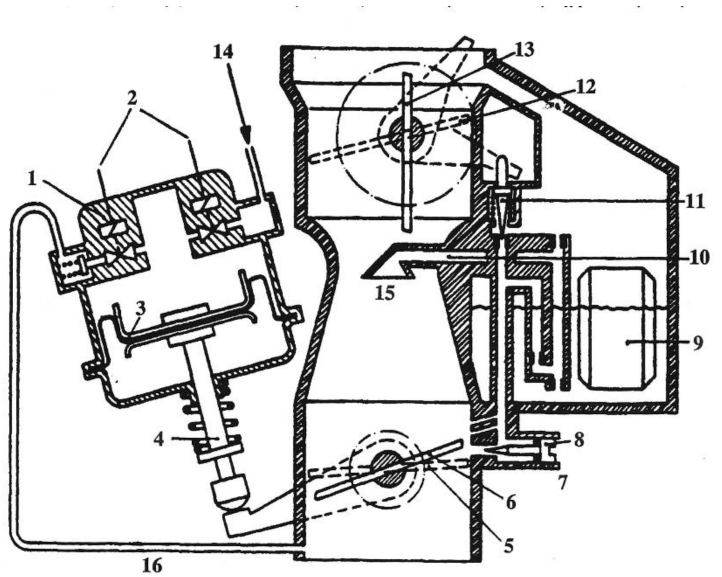


Figure (3.1.3) Ecotronic-ventilator.

1. electropneumatic mechanism
2. solenoid valve
3. membrane
4. butterfly valve push rod

5. throttle valve closed position.
6. throttle valve in vacuum operation
7. throttle screw
8. vacuum operating mixture.
9. float
10. central nozzle
11. air regulator
12. cold start position
13. pre-setting valve
14. atmospheric pressure
15. air direction
16. suction hose pressure

In the electro-pneumatic mechanism, a diaphragm moves the push rod of the throttle valve. The diaphragm is influenced by means of an electromagnetic valve, either by the pressure of the suction pipe or by the atmospheric pressure. The pre-regulating valve is regulated by a coil mounted and rotating in the magnetic field of a permanent magnet. By rotating the pre-regulating valve to the "closed" position, the mixture is enriched with fuel. Then a larger under pressure in the air direction occurs and more fuel exits the central nozzle system. With a cold start, the electromagnetic mechanism closes the pre-regulating valve, and the electro-pneumatic mechanism opens the regulating valve slightly. Since, in hot operation, in a cold suction pipe the fuel liquefies, the mixture must be enriched by maintaining a slight inclination of the pre-regulating valve.

The enrichment of the mixture is also necessary in accelerations and can be achieved by a small call of the pre-regulatory lock. In idling mode, the engine speed is automatically kept constant by the electro-pneumatic mechanism. At the same time, the pre-regulating valve and the air regulator are adjusted so that the mixture in vacuum mode saves fuel. In the boost mode (the vehicle running the engine), the throttle valve is closed by the electro-pneumatic mechanism for such a period until the vacuum mixture reaches atmospheric pressure. When there is no longer any fuel-mixture in the vacuum mode (the engine has no fuel), the number of revolutions is reduced to approximately 1100 rpm, the throttle is opened slightly, and the vacuum system is restored. Also, in the event of an engine shutdown, the throttle is closed, and spontaneous ignition is avoided. The Ecotronic - ventilator can also be integrated in the mixture formation of engines with catalytic converter. In the triadic catalyst, which minimizes the emissions of carbon monoxide, hydrocarbons and nitrogen oxides, the engine must be operated with an elemental mixture i.e. $\lambda=1.0$. A sensor, "the lambda sensor" , detects the oxygen in the exhaust gas between the engine and the catalyst. The measurement reaches the microprocessor, which adjusts the pre-regulatory lock via the electro-pneumatic mechanism so that the mixture composition is always equal to $\lambda=1.0$.

At the beginning of the century, airplane engines were operated by injection into the intake manifolds. The principle of high-pressure injection directly into the cylinder dates to 1930. The 4-stroke airplane engines under test were often involved in accidents, either because of low exhaust temperatures (freezing) or because of fuel overflow and ignition in the exhaust. The injection pumps used were those of diesel engines. The petrol was guided at the beginning of the piston stroke and injected after the exhaust pipe was closed, perpendicular to the air movement through the holes in the hubs or nozzles. A large valve overlap (the intake valve is opened before the ANS and the exhaust valve is closed after the ANS, so that both valves are open at the same time) makes it possible to achieve good sweeping (scavenging) of the cylinder of the residual exhaust gases without loss of fuel. The increase in power of the injection engine, compared to those with an exhaust, amounted to 17% with a simultaneous reduction in specific fuel consumption. In 1937 the production of the first gasoline-injected aircraft engines began and by the end of the war all aircraft engines were equipped with the injection system. By evenly distributing the mixture to all cylinders and by good flushing of the combustion chamber (a consequence of valve overlap), the limitations due to the "beat " of turbocharged airplane engines at high degrees of supercharging were avoided so that the piston power could be increased. The adjustment of the injection rate was independent of the pressure and temperature within the buffer-controlled suction duct. Also, by means of an altitude mechanism, the injection quantity was increased so that the external pressure dropped, because the drop in pressure left less residual exhaust gas in the cylinder. After the Second World War, in 1949, tests with petrol injection in the 2-stroke engines started again. The petrol was injected directly into the cylinder after the exhaust port was closed during compression. Despite good flushing, fuel losses were avoided. The piston power was increased, and the specific consumption was reduced. However, under full power delivery and low rpm, specific fuel consumption increases greatly, because the scavenging losses in the 2-stroke engine with exhaust are very high. The automotive industry then focused on direct injection under high pressure conditions. At a pressure of 50 to 100 bar, the fuel was injected into the nozzle in the cylinder during suction. To avoid the low injection volume problems of the multi-hole mechanism, the valve injection mechanism was developed. The risk of loss of scavenging (scavenging) of the 2-stroke engine, in the direct injection of fuel into the cylinder, is reduced in the 4-stroke engine. There is also the indirect injection mode, (Fig. (3.1.4.)). In indirect injection, the fuel is injected at low pressure into the intake manifold. The nozzle is constructed in such a way that the injected fuel reaches the cylinder space through the open intake valve.

Direct injection provides the following advantages: good external cooling of the cylinder by evaporation of the fuel droplets. Then, cylinder filling increases and the engine stroke limits are shifted to a higher compression ratio. Because, by design, the airflow is not interrupted at the intake manifold, cylinder filling increases. There can be no fuel return to the suction pipe. The engine can be adjusted with the amount of fuel mixture drawn up in such a way that a slightly flammable mixture is present in the igniter area and a poorer mixture in the remaining combustion area.

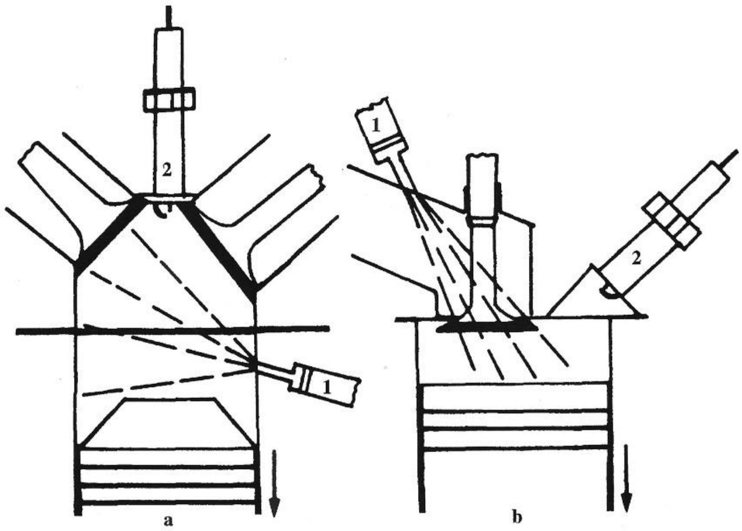


Figure (3.1.4) (a) direct injection, (b) indirect injection. 1) injection valve, 2) igniter.

The disadvantages of direct injection are that there is a risk of damaging the injection nozzle due to the high temperature, the construction of the injection nozzle makes the construction of the engine more complex, and, because of the high injection pressures, we have more demanding pumps and nozzles. These disadvantages are not encountered in injection in the suction pipe. The advantage of good internal cooling can be partly achieved. Today, most injection engines are equipped with suction pipe injection. Direct injection is still found in high-speed engines.

In summary, the reasons for injecting petrol are: better filling of the cylinder because the air flow is not obstructed, better valve coating with good flushing of the combustion chamber and reduction of overheating of the fuel mixture, shifting the "stroke " limits of the engine to higher compression ratios by providing equal amounts of mixture in all cylinders, with the valve overlap providing good scavenging of the residual exhaust gases and providing a rich mixture layer in the igniter and a poorer mixture layer in the rest of the combustion chamber. The above results in a higher ramming power. Also because the compression ratio can be increased less specific fuel consumption, there is no loss of fuel by purging the combustion chamber, the mixture can be made poorer without fear of afterburning, in the boost mode no fuel is injected and no acceleration adjustment is necessary to enrich the mixture, there is no risk of petrol backflow in the suction pipe when starting a cold engine, independence of engine operation in terms of altitude, precise adjustment of the quantitative petrol/air ratio at various loads and speeds.

There are still ventilators, because their construction is simpler, their price is lower and in terms of construction and installation they are simpler.

In suction duct injection, the nozzle injects either into the engine intake duct or through the open intake valve into the cylinder. Latest injection techniques give autonomy to each injection valve, with the advantage that some of the fuel in the cylinder is vaporized and internal cooling results in better filling and an increase in compression ratio. Its disadvantages are its more complex construction.

The Bosch K-Jetronic is shown in Fig. (3.1.5.); an electric fuel pump guides the petrol through the fuel pressure stabilizer and through the fuel filter to the fuel quantity distributor (the fuel pressure stabilizer keeps the fuel pressure at the right value for a long time when the engine is stopped, prevents vaporization and thus there are no problems with hot starting). From there it flows through the differential pressure valves in the intake pipes to the injection valves. The electronic fuel pump creates an overpressure of 4.7 bar, which is kept constant, via the pressure regulator, which returns the excess fuel back to the fuel storage. The spring-loaded injection valve is pushed with an overpressure of 3.3 bar and then remains open during operation, so that fuel is continuously injected into the intake manifold. The continuous injection gave the installation the name K-Jetronic (K=continuous). The determination of the fuel quantity for the injection valve is carried out at the fuel quantity distributor. There, the regulating piston leaves enough clearance in the regulating slot so that the correct amount of fuel (fuel volume in unit time) reaches the injection valve. For the fuel flow to depend only on the cross-section of the throttle slot (i.e., the position of the throttle piston), the flow velocity in the throttle slot must be constant in all operating conditions. This is achieved by means of differential pressure valves that maintain a constant pressure difference in the throttle slot equal to 0,1 bar.

The fuel quantity distributor will have one venturi and one differential pressure valve for each injection valve. The fuel flow is measured through the regulating piston independently of the air flow (air volume per unit time) directed to the cut-off disc through the air quantity meter. The cut-off disc is lifted by the air stream and this movement is transmitted by a lever arm to the regulating piston.

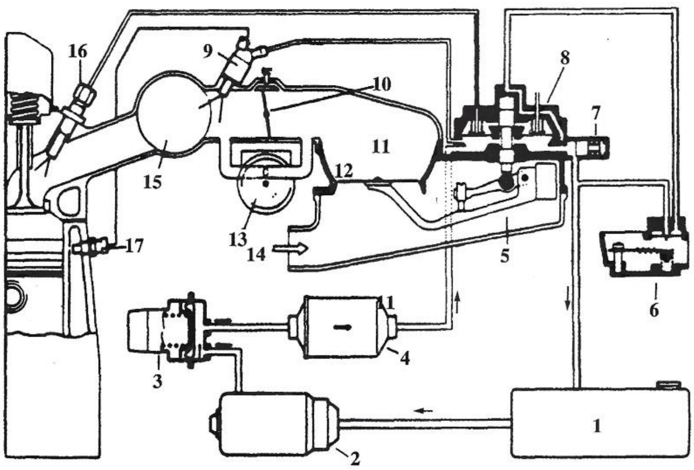
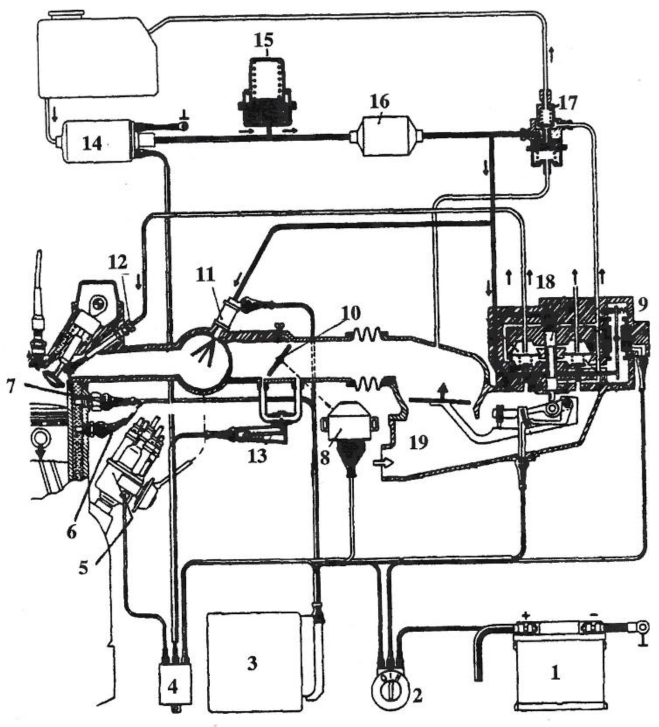


Figure (3.1.5) Bosch K-Jetronic.

1. fuel storage
2. electric fuel pump
3. pressure stabilizer
4. fuel filter
5. mixture regulator
6. hot service regulator
7. pressure regulator
8. fuel distributor
9. electric start valve
10. throttle valve
11. air director
12. cut-off disc

13. front air wire
14. air
15. concentric tube
16. injection valve
17. thermal timer

In this way, the air flow directly regulates the fuel flow. The force of the airflow on the cut-off disc balances the force on the throttle piston due to the throttle pressure. During the buffer cycle, fuel is drawn from a buffer hole in the fuel dispenser outside of the basic system operating cycle. In normal operation, in the buffer cycle, the overpressure is 3,7 bars. The level of the overpressure is adjusted by the operating regulator. With a cold start it is reduced to 0,5 bar. In this way, the air flow directly regulates the fuel flow. The force of the airflow on the cut-off disc balances the force on the throttle piston due to the throttle pressure. During the buffer cycle, fuel is drawn from a buffer hole in the fuel dispenser outside of the basic system operating cycle. In normal operation, in the buffer cycle, the overpressure is 3,7 bars. The level of the overpressure is adjusted by the operating regulator. With a cold start it is reduced to 0,5 bar. Lower buffer pressures indicate lower counter forces on the cutting disc. Consequently, greater airflow and the throttle piston provides a larger free slot cross-section to enrich the mixture with more fuel. In cold start, additional fuel is injected through the electric start valve into the main suction manifold. The thermal timer adjusts the opening and closing of the electric start valve as a function of the engine temperature. Because in cold start and closed thermal mode the engine friction is greater than in normal operation, the additional air drag provides a greater amount of mixture at the time. In hot operation it is closed via a bimetallic switch. Improvement of the K-Jetronic injection gave the development of the KE-Jetronic (figure(3.1.6)).



Figure(3.1.6) Bosch KE-Jetronic.

1. battery
2. starting switch
3. electrical regulating device
4. relay regulator
5. ignition distributors
6. engine temperature sensor
7. thermal timer
8. baffle switch
9. electrohydraulic pressure switch
10. throttle valve
11. cold start valve
12. injection valve
13. additional air latch
14. electric fuel pump
15. fuel pressure stabilizer
16. filter
17. pressure regulator
18. fuel distributor
19. air meter with cut-off disc and pressure gauge

KE-Jetronic is based on the K-Jetronic mechanism, but equipped with an electronic governor and sensors for engine temperature, position and movement of the throttle and cut-off disc. The pure fuel metering mechanism through the regulating piston is electronic, hence the addition of "=Electronisch", KE-Jetronic. To interfere with the electronic regulating device in the regulation of the fuel flow, the hot-working regulator was replaced by the electrohydraulic pressure switch, which receives commands from the electronic regulating device. The electro-hydraulic pressure switch allows a continuous flow of fuel from the main operating cycle to the lower chambers of the pressure valves.

The pressure exerted on the pressure gauge and at the same time on the size of the opening through which the fuel flows into the injection valves. Higher pressure in the lower chambers means that less fuel flows into the injection valves. In boost cut-off mode (vehicle running the engine, engine running without fuel), the electro-hydraulic pressure switch opens the system pressure so that the lower chambers are at high pressure and the hinge completely cuts off the flow of fuel to the injection valves. The electronic regulator takes care of the fuel enrichment of the mixture during cold start, hot start, acceleration, full power, and boost cut-off. It can also take on additional functions such as adjusting the mixture formation to varying pressure, air, or glow adjustment for triadic catalysts.

Figure (3.1.7) shows an electronic intake manifold injection mechanism, the Bosch D-Jetronic. The electric fuel pump supplies gasoline to the pressure lines of the solenoid injection valves. The pressure regulator keeps the power pressure constant at 3 bars. The injection timing and quantity are determined by the start-up and the duration of the opening of the injection valves. The injection valves inject one injection of fuel per operating cycle, i.e., in two rotations of the crankshaft, into the suction pipe. To simplify the construction, the injection valves are divided into two groups (four-cylinder 2 x 2, six-cylinder 2 x 3). Injection valves of each group are electronically connected and therefore open simultaneously. The principle of injection is determined by a cam acting on the contact switches on the distributor. The injection time is adjusted accordingly by the electronic control device. The injection valves remain open at the momentum of the flow for 2ms in vacuum mode and 8ms in full operation. The duration of the opening and thus the injection quantity is determined by the regulating device in conjunction with the absolute static pressure in the suction distributor behind the baffle, the number of engine revolutions and the temperature of the suction air.

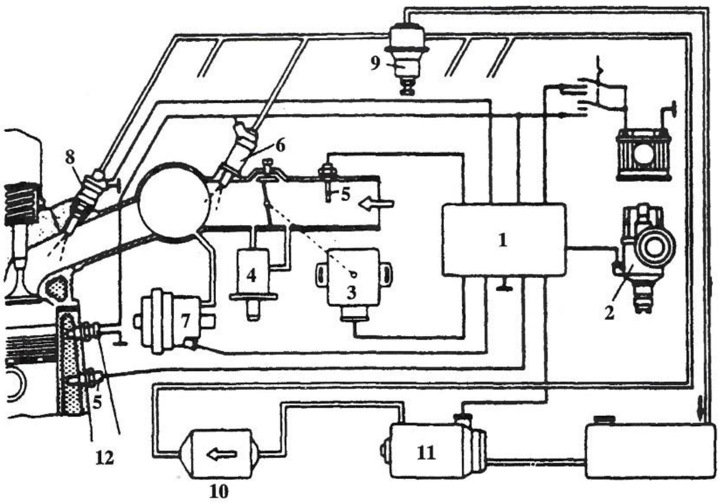


Figure (3.1.7) D-Jetronic Bosch.

1. electrical regulating device
2. ignition distributors
3. throttle switch
4. additional air latch
5. temperature sensor
6. cold start valve
7. pressure sensor
8. injection valve
9. fuel pressure regulator
10. filter
11. electric fuel pump
12. thermal timer

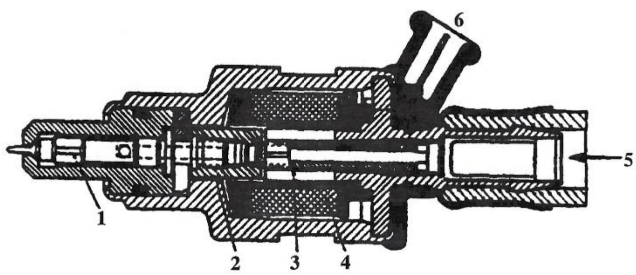


Figure (3.1.8) Electromagnetic injection valve.

1. valve nozzle
2. magnetic compartment
3. spring

4. magnetic winding
5. fuel
6. electrical circuit

The electronic tuning device also processes the injection amounts in the cold start, warm run, full power, acceleration and boost operating positions.

At cold start the mixture shall be enriched with petrol. A temperature sensor in the cooling water or in the cylinder head of an air-cooled engine transmits the temperature to the control device. Based on the detected temperature, the amount of fuel injected into the valves increases during pre-start and in addition fuel is injected into the suction distributor from the cold start valve.

During the warm running period a cold engine needs more amount of mixture in the idle mode to overcome the friction force than a hot engine. For this reason, the additional air drags which is regulated by the cooling water opens an additional duct in the throttle. More air in the suction distributor and an increase in air pressure result in an increase in the amount of injection. When the temperature of the cooling water increases the additional air latch closes the additional duct in the throttle again.

The enrichment in full operation is regulated by the pressure sensors. There is also an altitude correction because the change in density due to altitude is considered by the pressure sensors in the suction distributor and the temperature sensors.

Based on the experience with the D-Jetronic (D = Druckfulher = pressure sensor), Bosch has developed another electronic injection system, the L-Jetronic (L = Luftmengenmesser = air quantity meter). Figure (3.1.9) shows the L-Jetronic. It consists of an electronic regulation device, electromagnetic injection valves, an electric fuel pump and a pressure regulator. The pressure regulator adjusts the injection pressure to an overpressure of approximately 2,5 bar relative to the suction pressure. Unlike the D-Jetronic, all the injection valves inject fuel into the suction pipes at the same time, before the intake valves, which is a manufacturing simplification.

To form a uniform mixture in one operating cycle (two crankshaft revolutions), half the fuel is injected twice as much as twice as often. A matching of injection timing and crankshaft angle as in the D-Jetronic is not necessary. The injection timing is adjusted by the switch on the ignition distributor. Because the switch receives as many duty cycle signals as the engine cylinders (but only two signals are necessary for injection), the corresponding conversion is done in the adjuster. The duration of the injection valve opening, also called the injection time, determines the amount of fuel injected. It is calculated by the electronic control gear based on two main data. From the voltage signal measuring the quantity of air and the number of revolutions. The air quantity meter takes the place of the pressure sensor of the D-Jetronic.

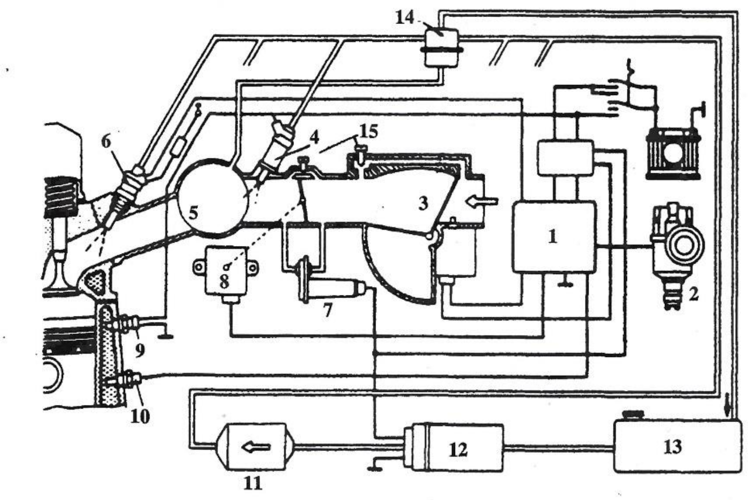


Figure (3.1.9) Bosch L-Jetronic.

1. electrical regulating device
2. ignition distributors
3. air counter
4. cold start valve
5. suction tube
6. injection valve
7. additional air latch
8. switch
9. thermal timer
10. temperature sensor
11. filter
12. fuel pump
13. fuel storage
14. pressure regulator
15. vacuum regulator

The L-Jetronic is simpler in construction than the D-Jetronic. Advantages of the L-Jetronic are the simpler and better determination of the fuel quantity measurement in the intake air quantity and the simpler possibility of driving the exhaust gases backwards (only the fresh air quantity is apportioned).

3.2 Emission of pollutants and catalyst installation in the Otto engine

The emission of pollutants depends on many factors, one of which is the formation of the mixture. Figure (3.2.1) shows the pollutants carbon monoxide, nitrogen oxides and hydrocarbons according to the air ratio λ.

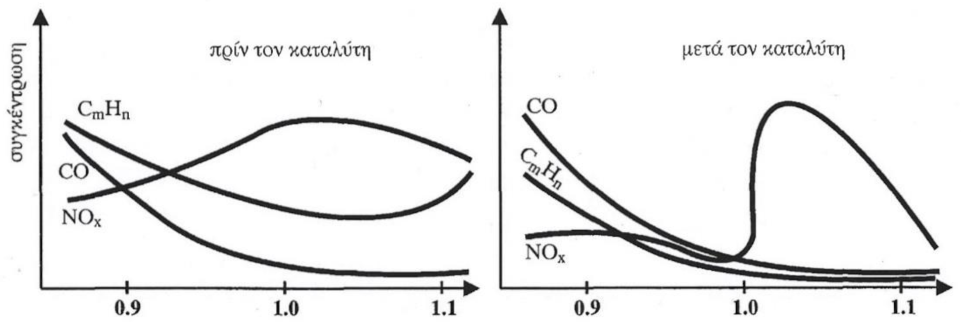


Figure (3.2.1) Harmful substances in petrol engine exhaust gases in recirculation with air ratio before and after catalyst.

The figure shows the large effect of the air ratio λ on the emission of pollutants. To reduce CO in exhaust gases, λ must be λ=1.1. In poor mixtures, to achieve good combustion quality, the composition of the mixture must be the same in each cylinder and from cycle to cycle and no differences should occur. This requirement is best achieved in a multistage degasser and even easier with spraying. By using a degasser during acceleration mode, CO increases due to the requirement to enrich the mixture. With both mixture formation systems (use of a deaerator, injection), there is no avoiding the rapid increase in EO at maximum power conditions unless the mixture enrichment requirement is waived in full power mode (= 0.9). This waiver would naturally mean a reduction in maximum engine power. In order not to exceed the carbon monoxide (CO) emission limit of 4.5% by volume in the exhaust gas when the engine is idling for a long period of time using a deaerator, an additional fuel supply mechanism is added. The idling of the engine is adjusted on the adjusting screw of the additional fuel supply mechanism (there is a change in the quantity of the idling mixture and not in the composition of the mixture) so that the quantity of CO in the exhaust gases remains unchanged.

The hydrocarbon emission λ=1.1 has a minimum value that increases again in a poorer operating range due to combustion residues, which can be reduced by good mixing of the mixture. The temperature of the combustion chamber walls also affects the hydrocarbon emission The walls should be less exposed to the temperature to avoid premature flame extinction. By reducing the surface area of the upper walls, the cooling effect is reduced.

The maximum emission of nitrogen oxides occurs at about λ = 1.05, because this is where we have the maximum temperature in the engine combustion chamber and the presence of sufficient oxygen. A large reduction in carbon dioxide emissions could only be achieved by a large increase in the air ratio. Nitrous oxide emissions are also reduced by driving the exhaust gases back into the engine. A certain amount of exhaust gas is added to the incoming air flow and to the hot mixture and its combustion temperature is reduced.

It turns out that with both mixture formation methods, degassing and injection, we can only reduce carbon monoxide and hydrocarbon emissions. When operating under partial power conditions, both mixture formation systems emit the same quantities of harmful substances. Under boost and acceleration operating conditions, the injection facility emits fewer pollutants than the degassing facility.

In the exhaust gases of the Otto engine, which contain the following harmful substances carbon monoxide, hydrocarbons, nitrogen oxides and lead compounds. Lead additives in petrol increase its resistance to impact. If the engines run on unleaded petrol, there are no more lead compounds. The three other harmful substances are indirectly reduced in the engine. The best way is to use a catalyst, which achieves a reduction in pollutant emissions of about 90% when the engine is operated at a ratio λ = 1.0 . Of the various catalyst installations, the three-way catalyst is described here. As their name implies, they simultaneously convert the three harmful substances CO, CmHn and NOx into non-poisonous substances: carbon dioxide (CO2) water (H2o) and nitrogen (N2). The following chemical reactions take place:

Oxidation $\mathrm{CO} + \frac{x1}{2} 02 \rightarrow CO2$

$$\mathrm{CmHn} + \left(\mathrm{m} + \frac{n}{4}\right) O2 \rightarrow mCO2 + \frac{\mathrm{n}}{2} + H2O$$

Reduction $\mathrm{NO} + \mathrm{CO} \rightarrow \frac{x1}{2} N2 + CO2$

Nitric oxide is reduced with the help of carbon monoxide to nitrogen with the simultaneous formation of carbon dioxide. The degree of conversion κ is given by the initial concentration of pollutants minus the final concentration of pollutants, from the initial concentration of pollutants. For 90% conversion of harmful substances we have κ = (1-0.1)/1 = 0.9, 90%.

To prevent reduction and oxidation from occurring simultaneously, the exhaust gases must contain a minimum of oxygen. Therefore, the engine must be operated with a metric mixture of λ = 1,0 and the deaerator as well as the injection system must be set to this value in every operating condition. The three-way catalytic converter only works in combination with an electronically adjustable mixture. The electronic adjustment device receives the complete mixture composition from a sensor in the exhaust stream. This adjustable device, 'the lambda sensor', is installed between the engine and the three-way catalytic converter. It provides an electrical voltage signal that is independent of the air ratio. The whole installation is built into the exhaust duct. Outside the platinum electrode flows low oxygen gas , while the inner platinum electrode is in contact with ambient air. The difference in oxygen concentration creates a voltage across the electrodes. The lambda sensor operates from 300 due to the solid electrolyte (drives the oxygen ions from this temperature and above). For the lambda sensor to quickly reach operation, it is equipped with an electrical resistor.

The three-way catalyst consists of a carrier body with an intermediate layer in which the specific catalyst material is transferred. A catalyst is a substance that increases the speed of a reaction by a chemical process rather than increasing the consumption. The carrier body is made in a chamber of sheet, ceramic or steel sheet material and consists of parallel channels through which the exhaust gas flows. The ceramic bearing contains 400 channels per square passage area with a wall thickness of approximately 0,2 mm. A layer of γ-Al2O3 is placed in the body of the bearing. This substance increases the active surface area to 10 to 25 m /g. In the intermediate layer there is a catalyst of platinum and rhodium in the amount of 1,5 to 2 g per liter of carrier. Platinum stops the oxidation reactions and with rhodium the reduction reactions.

The degree of conversion is independent of temperature. For solid conversion, the catalyst temperature must be above 300°C, but not exceed 900°C (reduction of catalyst lifetime). The location of the three-way catalyst construction in the exhaust must be selected in such a way that it operates in the desired temperature range under all engine operating conditions. Conversion is also exacerbated if particulates consisting of petrol, oil, etc. are added to the catalyst. Lead is particularly dangerous and therefore engines with catalyst installations should only use unleaded petrol.

3.3 Fuel Mixture Formation in the DIESEL Engine.

Diesel fuel is burnt in the diesel engine. Its boiling range at atmospheric pressure is between 200°C and 300°C and its kinematic viscosity is about 5x10-6 m2/s (20 °C). Comparison with the values for petrol (boiling range 40 and 200 °C and kinematic viscosity of about 0,5x10-6 m2/s (20 °C)) shows that degassing of diesel fuel is not possible. The fuel in the diesel engine is injected into the combustion chamber by a high-pressure injection pump at a pressure of several hundred bar. At the point of ignition, the fuel is injected at high velocity into the combustion chamber, becomes turbid in the air and an inhomogeneous mixture is formed. The small fuel droplets (with good nebulization their diameter is only a few μm , begin to evaporate in the hot air and burn. Evaporation and combustion occur on the outer surface, where heat and the presence of oxygen are combined. The combustion of the outer layer is very fast because the initial fuel molecules react immediately with oxygen. With a large total external droplet surface area, a lot of fuel is burned immediately and the pressure variation as well as the maximum pressure are large. The value of the pressure change must not exceed a certain limit to avoid mechanical overloading of the engine. The combustion of the outer surface of the fuel droplets increases the temperature inside. The high temperature and high pressure with the lack of oxygen lead to the breakdown of the fuel molecules.

Two categories of heat release processes during combustion in the diesel engine are considered. The large heat release resulting from the combustion of a large amount of fuel, before the top dead center, significantly increases the pressure, resulting in engine operating problems, while at the end of combustion, the heat release is significantly reduced by the decomposition reactions of the molecules with the risk of soot formation. Significantly more desirable behavior is the second category of heat release, in which combustion starts slowly and then increases in speed and although the same amount of heat is released as in the first case, combustion stops earlier. The following possibilities exist for regulating the combustion process, i.e., the release of heat.

The effect on the heat release from the injection is only applicable to low-speed engines, while in high-speed engines the large amount of injection cannot be adjusted.

Slowing down the combustion rate by accelerating the particle size effect, which can lead to the formation of soot.

The oxygen is mixed with the fuel in such a way that rapid combustion cannot be achieved, either the fuel is introduced as a liquid film into the relatively cold walls of the combustion chamber, or it is injected into the peripheral zone of the swirling air.

The methods of forming mixtures are classified into two categories:

α) Injection with air distribution (the fuel becomes cloudy by injection into the air).

b) Wall injection (fuel is injected in the form of a liquid film into the relatively cold walls of the combustion chamber).

In air distribution injection, the fuel is injected in such a way that it is distributed as evenly as possible before and during combustion throughout the air volume. There are two modes, direct injection, and ventilated injection (indirect), Fig. (3.3.1).

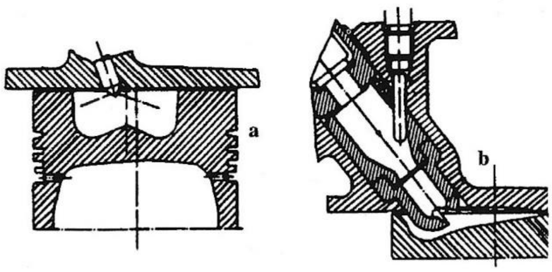


Figure (331) a. Direct injection, b. Indirect injection.

In direct injection the combustion chamber is integrated in the piston. The injection valve is located centrally in the cylinder head and injects fuel from a nozzle with up to 12 holes. The pressure at the nozzle outlet is 200 bar and the injection pressure at the pump is 1000 bar. At the intake, the air is rotated, and the swirling air causes the fuel to be distributed throughout the combustion chamber. The advantages of this mode are low specific fuel consumption, (because the flow losses during load change and heat losses are small), uniform and relatively low thermal stress on the combustion chamber walls and that no starting assistance is necessary.

The disadvantages of direct injection are that with increasing operating pressure and injection, the mechanical load is high, and the engine runs noisily. Also, if the engine is fuel sensitive, then the ignition delay must be short and finally, the intake must be configured to achieve air swirl even at low rpm. The latter leads especially in engines that have a wide rev range to poor cylinder filling at high revs. These disadvantages occur to a lesser extent in low-speed engines, and they must therefore be equipped with a direct injection installation. It is also currently prevalent in the engines of trucks that are swallowed for economy reasons. Direct injection operation up to a temperature of 15°C without the use of starting. Naturally, the engine starts to run with difficulty at lower temperatures emitting white smoke. White smoke appears when temperatures are below 250°C or when no ignition is available. Its exhaust gases contain small droplets (about 1 μm in diameter) consisting of hydrocarbons. At temperatures above 250°C it appears blue because the droplets in this temperature range are small.

In indirect fuel injection, the fuel is not injected directly into the combustion chamber (Fig. (3.3.1)), but into the pre-chamber or a swirl chamber. The volume of the chamber is approximately 25% to 60% of the final combustion chamber. The injection valve is located inside the chamber and injects the fuel at a lower pressure than direct injection through a knob nozzle (outlet pressure about 100 bar). In the swirl chamber, the fuel distribution is supported by the swirling air, which is created during compression when the air is compressed through the tangential expansion duct in the chamber. There is not much air movement in the antechamber. Both chambers have high temperatures that reduce the ignition delay. Due to the high temperature and lack of oxygen, many molecular decomposition reactions take place. Molecular decomposition and lack of oxygen lead to slow pre-combustion in the chamber. The increase in pressure in the chamber drives its contents at high speed into the main chamber. This is where the rest of the combustion takes place. In the main combustion chamber strong turbulence occurs and the dissolved fuel burns without the formation of smoke (soot).

An advantage of indirect injection is that the high chamber temperature contributes to a short combustion delay. The engine is not fuel sensitive. The pressure rise and injection pressure remain low because decomposition and lack of oxygen lead to a more stratified combustion. The engine runs more smoothly than with direct injection. The injection pressure is lower, and the injection pump and nozzle are less stressed. The nozzle hub is self-cleaning, so the nozzle is not damaged. The filling of the cylinder at high rpm is better.

A disadvantage of indirect injection is the higher specific fuel consumption. Due to the additional chamber, high fluidity losses and heat losses due to the larger combustion chamber surfaces. Thermal stress is particularly high at the chamber orifices and in the piston body. During start-up, auxiliary means (lamps) are necessary. The chamber engine is suitable for multi-fuel engines, with hydrocarbon fuel in the boiling range from 40 to 450 °C, also oils and alcohol.

The method of forming a wall-partitioned injection mixture is the way in which the one-hole (formerly two-hole) nozzle injects the fuel in the direction of air rotation on the outer spherical surface of the piston, (M-mode). The swirling air retains the liquid film on the walls (thickness at full operation about 15 μm), A small piece of fuel is detached from the main jet, meets the air and ignites, initiating combustion. The swirling air evaporates the fuel in layers from the oily, relatively cool piston walls. The gasoline vapors mix with the air and ignite. Because the liquid fuel remains relatively cold on the piston walls, decomposition reactions take place and soot formation is prevented. Combustion then starts slowly. The increase in turbulence accelerates evaporation and combustion. Heat release is desirable, pressure rise, and peak pressure remain low, and the duration of combustion is short.

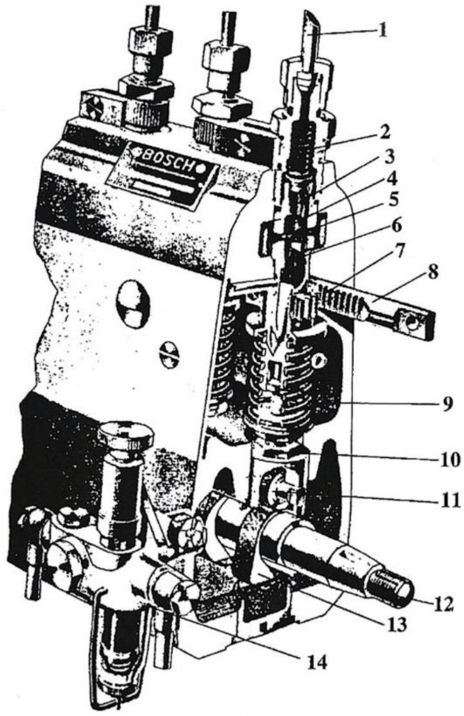
Further development of mixture formation with the M method led to the EM method, where the average effective pressure was increased with a simultaneous reduction in specific fuel consumption and ignition pressure. Combustion occurs with both mixture formation methods and with fuels with a long ignition delay and therefore both methods are suitable for long combustion engines. Difficulties are encountered in the combustion of Super petrol. The problem is overcome by the construction of an auxiliary ignitor. The new way was called the FM method. From the pure Diesel engine, with the construction of the ignitor, we were led to the construction of a mixed engine. The M-method of mixture formation is a clear advantage of direct and indirect injection. The engine runs smoothly, like a cam engine, and the specific fuel consumption is as low as in direct injection distribution in air.

In the diesel engine, the active pressure of the medium and thus the power is not regulated as in the Otto engine by filling the cylinder, but by continuously filling the air by injecting a quantity of fuel. Consequently, the diesel engine operates in the partial power range with a high air ratio λ. The compression ratio e must be chosen so high that even in the partial power range and especially in hot operation ignition and combustion can take place even with a cold engine. While in full power operation there is no problem, the necessary high temperature for a good combustion process in the partial power range is achieved by reversing the burners. With this method, the exhaust gases are led into the intake duct through a connecting pipe. This is achieved by means of a temperature sensor so that, when the engine temperature drops, more exhaust gases are mixed in the intake manifold. During combustion, there is no shortage of oxygen because the amount of fuel injected in partial power mode is small.

Rear exhaust flow offers the following advantages: the compression must be chosen high enough to allow the engine to run safely at full power, thus keeping the ignition pressure low and reducing the engine load. By increasing the intake temperature during partial power operation, the ignition delay is reduced and the engine's sensitivity to fuel is increased. It can then burn many different fuels. The exhaust gases contain less nitrogen oxides, but naturally the emissions of hydrocarbons, carbon monoxide and particulates increase. And in the Otto engine the reverse exhaust gas flow is used to reduce nitrogen oxides.

The injection system in the diesel engine consists of the injection pump, injectors, and injection valves. The injection pump must supply fuel at high pressure (up to 1000 bar), store the injection quantity regardless of the engine load, inject the fuel at the right time and preset. Figure (3.3.2) shows an injection pump. The camshaft driven by the engine pushes the injection piston with a ball drive mechanism. The number of revolutions that a four-stroke engine has is perhaps half that of a two-stroke engine with the full number of revolutions of the propeller shaft. The injection piston provides a constant injection. Storage of the injection quantity is achieved by rotating the injection piston around it along the shaft.

Fuel delivery begins when the upper end of the piston is exposed to the intake manifold. The flow rate and flow end are determined by the side control. If the start of flow is fixed, the end of flow with the injected quantity. The adjustment of the quantity, i.e., the pistons, is done by the adjustment rule. Between the start of flow and the start of injection, there is a fixed time. The start of combustion is detected by adding the ignition delay. The time difference between the start of flow and the start of combustion is independent of the number of revolutions. As the rpm increases, the crank angle between the start of flow and combustion increases, i.e., combustion starts later. Therefore, in engines running at high range and high rpm, to prevent combustion from deteriorating, the start of the flow is regulated and therefore the injection pump is equipped with an injection controller.



1.Pressure pipeline mechanism — 6.injetion piston — 11.spherical drive

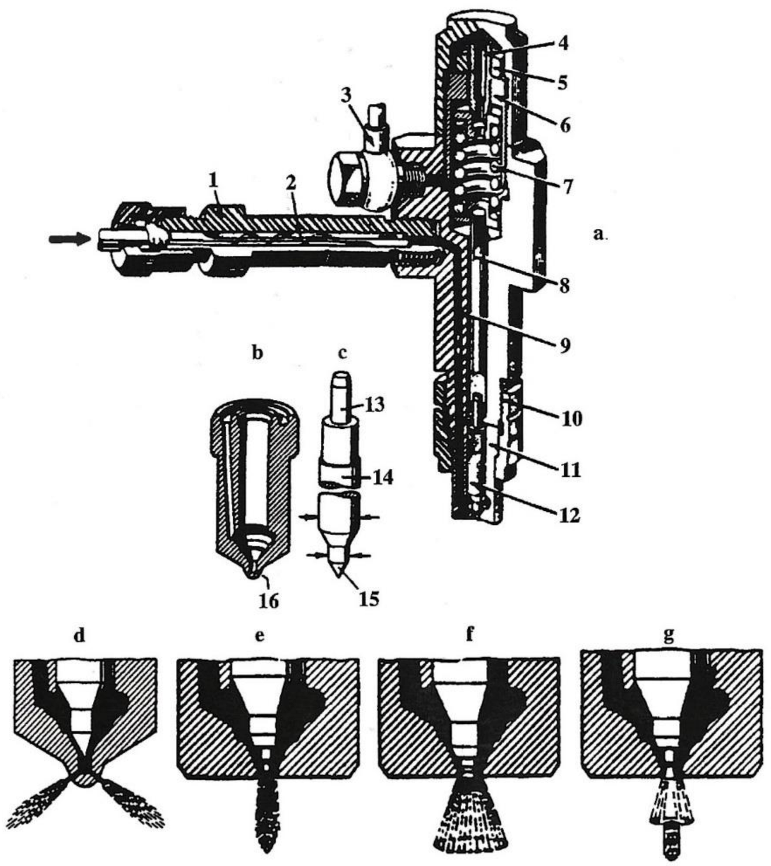
1.Pressure pipeline mechanism	6.injetion piston	11.spherical drive
2.pressure valve	7.toothed rim	12.camshaft.
3.combustion chamber	8.regulatory rule	13.cam
4.cylinder	9.piston spring	14.fuel supply pump
5.suction chamber	10. adjusting screw	

Figure (3.3.2) Bosch injection pump.

The pump components, i.e., the injection pistons and injection cylinders are assembled and work together with great precision. In pumps for multi-combustion engines, a notch in the injection cylinder prevents fuel from entering the pump cartridge. The pressure valve (check valve) is located at the outlet of the pressure pump chamber.

The pressure valve serves two purposes: the fuel of the injection piston after the end of injection is not sucked into the injection pipe and it reduces the pressure resulting from the injection in the pipe, achieving closure of the pressure valve.

In cooperation with the injection pump, there is also the injection dispensing pump, which feeds the fuel and distributes it to the injection valves (Fig. (3.3.3)). The feed pump, which is integrated into the pumps' operating shaft, pushes the fuel into the pump's suction chamber (Fig. (3.3.4)). From there it flows to the high pressure, as well as to the pump's regulation section.



1.Supply connection	9.fuel hole
2.filters	10.nozzle connection
3.return fuel outlet	11.nozzle
4.regulatory stem	12.needl
5.rail adjustment	13.pressure button
6.spring cover	14.needle body
7.spring	15.mounting surface
8.stem	16.injection hole

Figure (3.3.3.) a:Bosch injection valve, b:body, c:needle d:open hole nozzle e:open hub nozzle f:cylindrical end of knob, g:tapered knob end.

A special dispensing piston provides high pressure and fuel regulation. From the high-pressure section, it pushes fuel through the pump outlet to the injection valves. The adjustment achieves the correct dosage (storage) of the injection quantity. The distributor piston performs three tasks, it supplies high pressure fuel to the injection valves, it supplies fuel to the regulator, the regulator slides and distributes the fuel to the various pump outlets. The distributor piston is actuated by the pumps' operating shaft.

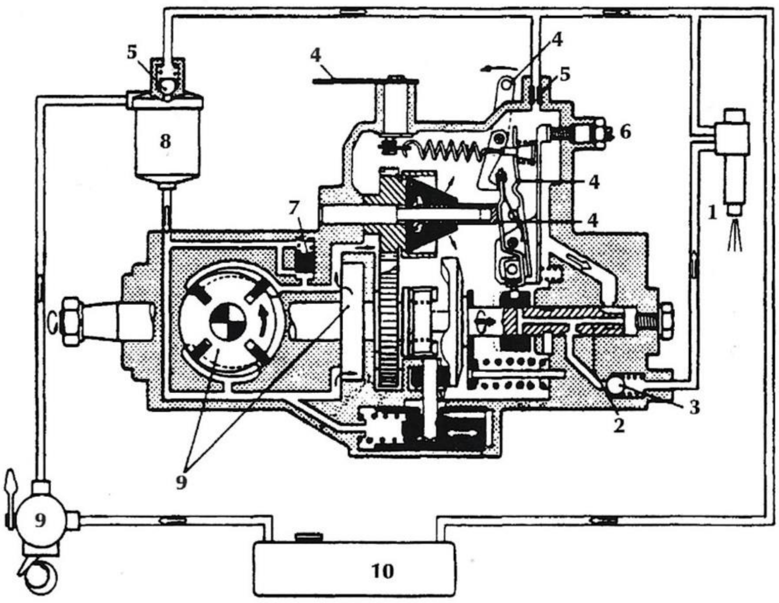


Figure (3.3.4) Distribution - mechanical spray pump regulator (Bosch), 1 nozzle. 2 dispensing pump, 3 dispensing valve, 4 operating level control, 5 overflow valve, 6 full load adjusting screw, 7 pressure adjusting valve, 8 filter, 9 feed pump, 10 fuel tank.

The injector connects the injection pump to the injection valve. It is made of steel pipe with an external diameter of 6 to 8 mm and a wall thickness of between 1,5 and 2,5 mm. For the same timing and the same injection quantities, the pipes must have the same length in all cylinders, without neglecting the effect of the pressure surge and the compression of the fuel.

Injection nozzles are bore nozzles and node nozzles. Hole nozzles are manufactured in direct injection engines and have up to 12 holes. The diameter of the holes starts from 0,2 mm. The pressure in the bore of the nozzle ranges between 150 and 250 bar. Node nozzles are used in engines with a pre-chamber. The pressure in the nozzle bore is lower and ranges from 80 to 125 bar.

3.4 Pollutant Emissions in the DIESEL Engine

The diesel engine contains carbon monoxide, nitrogen oxides, hydrocarbons, aldehydes, Sulphur dioxide additives and soot. The diesel engine has fewer harmful substances in the exhaust gases than the Otto engine because it runs on a higher proportion of air.

Carbon monoxide is only 0.02 to 0.12% in the diesel engine, as opposed to 1 to 4% in the Otto engine. The formation of nitrogen oxides depends on the temperature and the concentration of oxygen during combustion. At lower combustion chamber temperatures in the diesel engine, less nitrogen oxides are produced than in the Otto engine. Only aldehydes are found in larger quantities in the diesel engine. According to DIN requirements, the permissible Sulphur content in petrol is 0,1 % and in diesel 0,5 %. In heavy oils burned in large diesel engines, the Sulphur content can reach up to 5 % and vice versa, and the concentration of Sulphur dioxide in the exhaust gases increases.

In the diesel engine, the problem is the formation of soot. If a very small part of the fuel is not completely burnt, it colors the exhaust gases with the familiar dull color. If the amount of soot in the exhaust gas exceeds 0.15 g/m, the color becomes visible. The carbon itself is not harmful to health, but the carcinogen 3,4-benzopyrene in the soot is. The cause of soot formation lies in the principle of decomposition of molecules during combustion. Therefore, it must be avoided by good distribution of the injection to the walls of the combustion chamber or by seeking good combustion of the soot in the engine by achieving good turbulence. Also, fuel additives can help against soot formation. In addition, it is possible to remove soot from the exhaust gases by means of ceramic soot filters, which are integrated into the exhaust duct.

When researching the ways of mixture formation and combustion to achieve low emission of harmful substances and at the same time low fuel consumption, various ways of stratified feeding were found. According to those working with external ignition, the mixture formation is adjusted so that the igniter has a rich fuel mixture (λ = 0,6 to 0,9) and the rest of the combustion chamber has a much poorer mixture. The air ratio in the entire mixture is 2 or even 3, where an Otto engine would not work. Where it is known that for air ratio λ values greater than 1,3, the formation of nitrogen oxides and carbon monoxide is reduced. Above the value λ = 1,3 is also the operating range of the stratified feed engine. The stratified feed engine emits only a small number of hydrocarbons. The specific fuel consumption is lower than that of the Otto engine. Of course, the stratified feed engine does not reach the pulse power of the Otto engine because the mixture is poorer, and its heating power is lower.

Figure (3.4.1) shows a subdivided combustion chamber. A small antechamber is connected via an expansion tube to the main combustion chamber. Fuel is injected into the antechamber and ignited by the igniter. The flame spreads through the expansion tube and ignites the poor mixture in the main combustion chamber. The stratified fuel engine with a subdivided combustion chamber receives the mixture, as in the Otto engine, from the ventilator or by injection into the intake manifold. The stratified fuel engine has characteristics of the Otto engine as well as the Diesel engine, e.g. igniter and pre-chamber. For this reason it belongs to the hybrid (mixed) production engines.

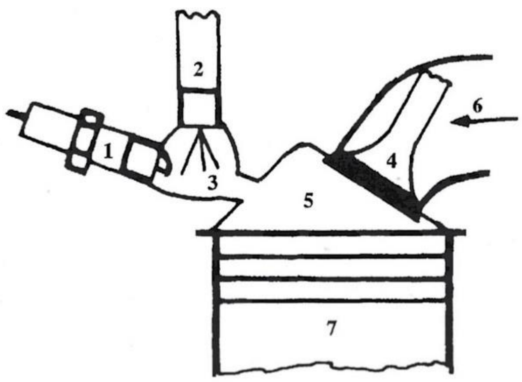


Figure (3.4.1) Stratified fuel engine with subdivided combustion chamber, 1 igniter, 2 injection nozzle, 3 pre-chamber, 4 valve, 5 main chamber, 6 intake, 7 piston.

After common and super gasoline, catalytic converters are in danger of becoming obsolete, which already have a problem of catalyst replacement in Greece, resulting in excessive emissions. After the year 2000, there is expected to be an "explosion" in the market for alternative energy car models worldwide.

An alternative fuel to petrol, methyl ester from the radish plant, commonly known as "biodiesel", is already in use in Germany and has the same performance as diesel but emits a quarter of the exhaust emissions. There are already many biodiesel filling stations, and car manufacturers have created modified models that run on biodiesel. German taxis are promoting the widespread use of biodiesel, while farmers receive subsidies from the EEC and the German state for growing radish, a renewable energy source from which biodiesel is obtained by simple distillation.

Mexico City, which, because of its altitude, has the worst air pollution in the world, will get 2000 taxis, which will not only not pollute the air, but clean it. These are the Zero Pollution Urban Taxis, an invention of the French engineer Gi Negr, who has put a vehicle into service that follows the following operating procedure: 300 liter of compressed air give the vehicle a range of 200 kilometers. Specially designed for urban traffic, where starting and stopping cars multiplies pollution, Zero Pollution taxis can be filled either in one minute at specially equipped petrol stations or in 4 hours anywhere with an electric compressor. Its engine, in addition to the combustion chamber, includes an antechamber that compresses the air at high temperatures. A small amount of this compressed air is then introduced into the combustion chamber at ambient temperature, increasing the pressure that sets the engine cylinders in motion. There is also a carbon filter system that absorbs the polluted air when the vehicle brakes, resulting in an exhaust air that is cleaner than the ambient air.

Recently, American scientists have discovered a new technique for converting traditional fuels such as oil into electricity. The operating principle of the new car is based on the ability of petrol and fuel to break down under the influence of hydrogen, a product which in electric form is fed into cellular engines that produce zero to minimal exhaust emissions. According to his estimate, the Electrochemical Car will be ready for the market in 5 years or perhaps sooner.

Also, surprises are expected from Japan, Toyota recently announced that a car will soon be on the market that will convert gasoline into electricity to charge the battery, but not directly to the engine.

In all cases, the petrol station remains off and the prospect of charging the batteries at home or in energy conversions will be done inside the car itself.

CHAPTER 4

4 Mechanical Analysis and Weighing

4.1 Kinematics and Dynamics of Internal Combustion Engines

Otto and Diesel engines are found as reciprocating internal combustion engines and rotary internal combustion engines. These two characteristics indicate the way in which the piston moves. In the first category (Fig. (4.1.1)) the piston moves between two extreme points, the dead centers (top and bottom). The distance between the dead points is called the piston travel.

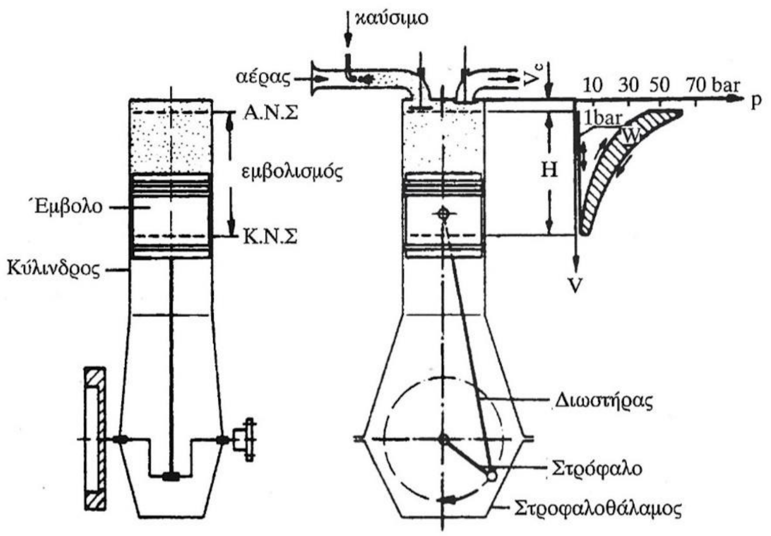


Figure (4.1.1) Engine piston

The driving mechanism of the reciprocating engine consists of the piston, the court, and the crankshaft. The movement of the piston is guided by the cylinder. The crankshaft is supported on its bearings. The change of energy due to the pressures into mechanical work takes place at the top of the piston. The force created by this change is transmitted from the piston to the crankshaft and results in torque. The purpose of the crankshaft is to convert reciprocating motion into rotary motion.

Figure (4.1.2) shows a schematic representation of a rotary internal combustion engine.

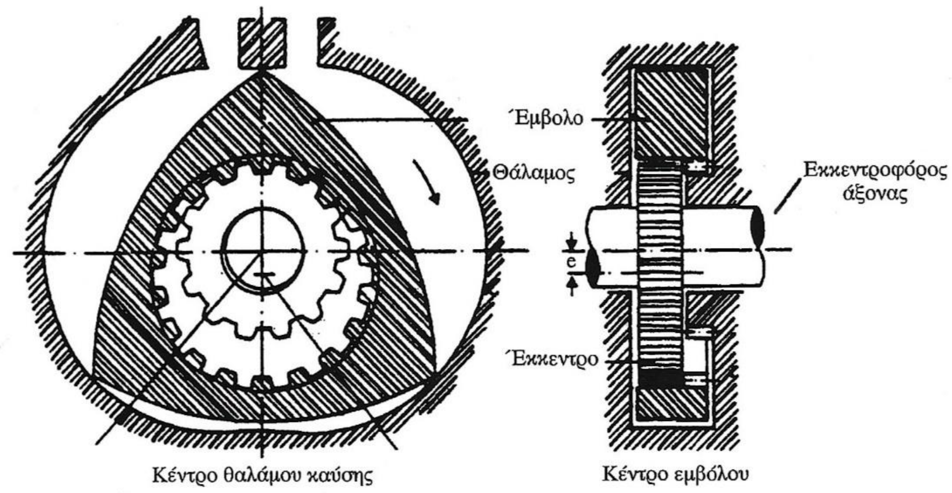


Figure (4.1.2) Rotary engine.

The driving mechanism of the rotary internal combustion engine consists of the piston and the camshaft. The piston is moved in the right orbit by the camshaft and the two gears. The camshaft bearing lives in the engine body. On the three frontal surfaces of the piston, the conversion of energy due to the pressures into mechanical work is completed. The force created during this conversion pushes the cam, which in turn creates torque on the camshaft. In rotary piston internal combustion engines there are no dead spots because the motion is continuous. The rotary piston internal combustion engine has the advantage of the piston rod that it does not have to be converted back to linear rotary motion and no free mass forces occur. However, the question arises as to why these engines have not become prevalent in recent years, since they have so many advantages. Rotary piston engines have been manufactured for many years. However, all these machines have not been preferred because the sealing of the piston, which is stressed by high-temperature and high-pressure gases, has been weak for some years. In 1954, Felix Wankel succeeded for the first time in building an ICE rotary piston engine, which was free of sealing problems. This engine was built by NSU in Neckarsulm, Germany and was first operated in 1957. After several years of intensive research and development, it was ready for proper and reliable operation and today it is widely used in both industry and transport. The ICE rotary engine is named after its inventor and is called the Wankel engine. This engine works by the process of four-stroke engines. The inlet and outlet diodes for filling are regulated by the piston. One rotation of the piston corresponds to three rotations of the camshaft. Since three operating cycles occur in one rotation of the piston, it is obvious that one operating cycle corresponds to each rotation of the camshaft. To obtain the displacement of the Wankel engine, we add the displacement of the two combustion chambers. This can be understood by comparing it to a four-stroke piston engine, by corresponding one rotation of the camshaft to one rotation of the crankshaft. For one duty cycle to correspond to one crankshaft rotation in a four-stroke piston engine, the engine must have two cylinders and the total displacement must be equal to twice the cylinder displacement.

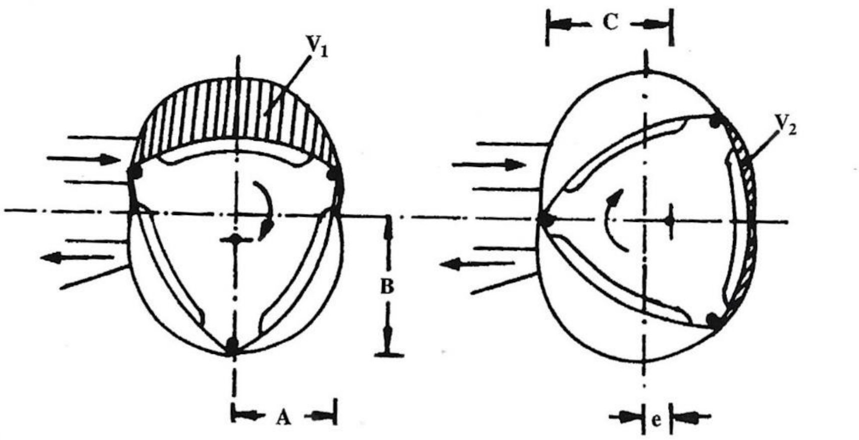


Figure (4.1.3) Piston chamber space of a rotary piston engine.

The cubic capacity of a chamber is equal to the difference between the maximum chamber minus the minimum, (Fig. (4.1.3):

$$Vh = V1 - V2$$

The calculation ratio of the piston chamber is Vh = 5.20eCπ, where e is the eccentricity, C is the height of the top and π is the width of the piston. The calculation of the piston chamber can also be done using the two semi-axes of the chamber:

$$Vh = 1.3\pi(B^2 - A^2)$$

major axle:

$$B = C + e$$

minor axis:

$$D = C - e$$

The Wankel engine was until now manufactured as an Otto engine, because the configuration of the combustion chamber surface was not suitable for a Diesel engine. Wankel engines are manufactured for a wide range of power. Larger engines are built to have up to four rotating pistons mounted on a single camshaft. Large engines are water-cooled, while smaller engines are air-cooled. The Wankel engine is characterized by its low operating noise (low reciprocating mass forces), small volume and low weight.

In the reciprocating piston engine, the piston performs an uneven motion between the two dead centers (sometimes accelerating and sometimes decelerating). The crankshaft, on the other hand, rotates at a constant angular speed. The movement of the connecting rod is determined by the movements of the piston and the crankshaft. The piston is limited by its upper possible position (Top Dead Center, TDC) and the lower possible position (Bottom Dead Center, BDC), The distance between these points is called the piston stroke. This piston stroke s is expressed as a function of the piston angle α (Fig. (4.1.4.):

$$s = 1 + r - (Icos\beta + cos\alpha)$$

The angle b of the connecting rod by means of the relation:

$$l\ sin\beta = r\ sin\alpha$$

and entering the slope ratio of the connecting rod:

$$\lambda = \frac{r}{1}$$

we have:

$$sin\beta = \lambda\ sin\alpha$$

And

$$cos\beta = \sqrt{1 - \lambda^2 * \sin^2(\alpha)}$$

Thus, we arrive at the type of piston stroke.

$$s = r * (1 - \cos(\alpha)) + 1 * \sqrt{1 - \lambda^2 * \sin^2(\alpha)}$$

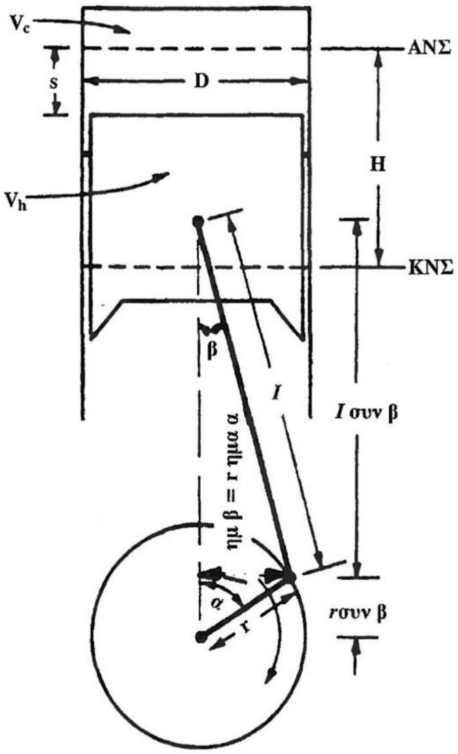


Figure (4.1.4) Piston path.

But mathematical operations are difficult because of the root cause. For this reason, an approximate equation is used, the accuracy of which serves us in most cases in engineering applications. The approximate equation is obtained by converting the root to a series and truncating the second order terms and above. Series :

$$y(x) = y(0) + y'\frac{(0)*x}{1!} + \ldots.$$

With:

$$\lambda^2 * \sin^2(a) = x$$

We have:

$$sqrt1 - \lambda^2 * \sin^2(a) = y$$

So:

$$y = \sqrt{1-x}$$

According to the series:

$$y = 1 - \frac{0.5 * x}{1!}$$

the approximate equation for the root is obtained:

$$\sqrt{\left((1 - \lambda^2 * \sin^2(\alpha))\right)} = 1 - \frac{\lambda^2}{2} * \sin^2(\alpha)$$

The expression of the stroke of the piston is given by the relation:

$$s = r(1 - \cos(\alpha) + \tfrac{\lambda}{2} * \sin^2(\alpha)) \qquad (4.1.2)$$

Another very important quantity is that of the speed of the piston. The easiest to calculate mathematically is the average piston speed, which is very often used as a benchmark. The average piston speed is defined by the distance-time relationship. The path is taken as the double path of the piston 2H. This stroke corresponds to one rotation of the crankshaft in 1/n time. The quantity n is the number of revolutions per unit time. The formula for the average speed of the piston is:

$$um = 2Hn$$

In practice, however, we use the formula:

$$um = \frac{\mathrm{Hn}}{30}$$

H (m), n (1/min), Um (m/s).

In some cases, we are also interested in the instantaneous (real) velocity of the piston, which is obtained from the derivative of the path with time as a function of the crank angle:

$$u = \frac{ds}{d\alpha} * (\frac{d\alpha}{dt})$$

The expression da/dt is the angular velocity ω of the crank. Starting from relation (4.1.2), we also arrive at an approximate equation for the piston speed. Its accuracy is sufficient for most cases used in engineering applications. By:

$$\frac{ds}{da} = r * (\sin(\alpha) + \lambda sin(\alpha) * \cos(a)) = r * (sin\alpha + \frac{\lambda}{2} * \sin(2\alpha))$$

Finally:

$$v = \omega r * (\sin(a) + \tfrac{\lambda}{2} * \sin(2\alpha)) \qquad (4.1.3)$$

The acceleration of the piston is obtained in a similar way if we derive the speed of the piston:

$$\gamma = \tfrac{du}{dt} = \tfrac{dcd\alpha}{d\alpha dt} = \omega^2 r(\cos(a) + \lambda cos(2\mathrm{a})) \qquad (4.1.4)$$

For machines with a large number of operating speeds (high speeds), the accuracy given by the above equation is not sufficient. Then, a derivation must be made, either in the exact equation of the piston stroke (relation 4.1.1), or by developing the root in exponential order with more than two factors.

The piston of a rotary piston engine rotates around the cam (Fig. 4.1.2). On the large non-fixed gear, it rolls internally on the small fixed gear. The cam acts so that the centre of gravity of the piston and the centre of gravity of the fixed gear (centre of the chamber) are always at the same distance e (e = eccentricity). The cooperation of the two gears gives a peculiar unfolding movement. The combination of cam and gearing results in a piston movement in which the piston ends describe a curve which is described as a peripheral or epicyclic movement. Like the peripheral, the epicyclic is designed geometrically. Strict geometric construction is, among other things, time consuming. The combustion chamber of a rotary piston engine is most succinctly designed by the design procedure of Figure (4.15). For this design method a white and transparent sheet of paper is necessary. On the white paper the base circle with radius p = 2 cm is drawn and on the transparent paper the cylinder with radius p = 3 cm is drawn. To achieve the unfolding motion, the two circles are divided into equal arc segments and the intersections of the radii with the circumference of the circle are numbered from 1 to 5. To find the geometric centre of the rolling circle, an equilateral triangle is drawn in which the distances from the centre to the edges are 7,5 cm. This triangle represents the piston. Thus, we can draw the combustion chamber, i.e. the peripheral motion. In figure (4.1.5), the cylinder is allowed to roll on a base circle so that the equal points and radii of the circles coincide. The edges of the equilateral triangle are marked at each position t by transparent paper on the white paper. The curve passing through these marks is the desired orbital motion. The actual perimeter of the chamber deviates slightly from the peripheral motion. This is because the piston has no sharp edges, so the gases cannot move from one side of the piston to the other. The overlap curve is the actual perimeter of the chamber. Then the overlap and the peripheral motion have the same meaning.

The following useful conclusions can be drawn from the rotary motion construction with regard to the motion of rotary piston engines: during the winding motion, the distance between the centre of the combustion chamber and the centre of the piston always remains constant and equal to the distance e. One complete rotation of the piston corresponds to three basic rotations of the camshaft, so that the centre of gravity of the piston, which coincides with the centre of the camshaft, rotates three times at the same time.

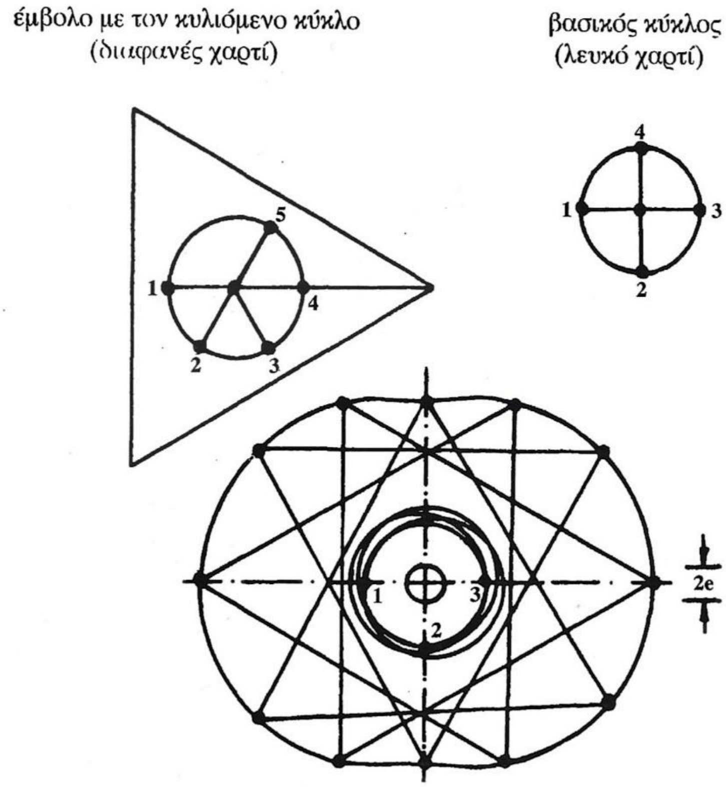


Figure (4.1.5) Design of peri trochoid motion.

The "gas forces" come from the pressure of a gas on a surface. The pressure of the working medium is applied to the upper part of the piston of a reciprocating piston and the atmospheric pressure to the lower part. The force due to the gases is given by the formula:

$$F = E * Poverpressure \qquad (4.1.5)$$

E is the projection of the piston surface onto a surface perpendicular to the piston axis and p overpressure is the operating overpressure through them.

The path of the forces generated by the 'gas forces' in the reciprocating machine proves that no external force is exerted and that all forces are balanced within the machine.

From the uneven movement of the masses, "mass forces" develop. These forces are found in both reciprocating and rotary reciprocating machines. The forces in rotary machines are fully balanced, whereas in reciprocating machines they are only partially balanced. The balancing of the 'mass forces' is done by opposing forces, so that the action of the mass forces is not transferred outside the machine. The 'mass forces' vary periodically and are divided into rotational and reciprocating forces. Rotational forces are known as centrifugal forces and are generated by the uniform cyclical movements of the masses. To calculate the rotational forces of the masses, the following relation is used:

$$Fr = mrr\omega^2 \qquad (4.1.6)$$

mr is the rotating mass, r is the distance of the center of gravity of the mass from the point of rotation and as the angular velocity.

The reciprocating mass forces are created by the non-uniform motion of the piston. The calculation shall be made by means of the following relationship:

$$F_{\pi\alpha\lambda} = m_{\pi\alpha\lambda}\gamma \qquad (4.1.7)$$

Where mpal is the reciprocating mass and γ is the acceleration. If we substitute the acceleration of the piston in relation (4.1.4), then the force of the tidal masses is calculated:

$$F_{\pi\alpha\lambda} = m_{\pi\alpha\lambda} r\omega^2(\cos(a) + \lambda cos(2\text{a})) \qquad (4.1.8)$$

In the rotary piston engine only rotational mass forces occur. These come from the masses of the piston and the camshaft, and both masses rotate as the camshaft rotates at a distance e from the centre of the combustion chamber. The relationship for the rotating mass force in rotary piston engines is:

$$Fr = (m_{\varepsilon\mu\beta\acute{o}\lambda o\upsilon} + m_{\varepsilon\kappa\kappa\acute{\varepsilon}\nu\tau\rho o\upsilon})e\omega^2 \qquad (4.1.9)$$

In reciprocating engines, calculating the forces of the rotating masses is more difficult. The rotating masses include the crankshaft, the crankshaft knobs and the rotating part of the connecting rod (Fig. (4.1.6). The rotating part of the connecting rod is calculated on the basis of the following reasoning. The motion of the centre of gravity of the connecting rod cannot be explained by a simple reasoning. The connecting rod head (the part of the connecting rod that is connected to the piston pin) has a reciprocating motion. The lower part of the connecting rod (the part of the connecting rod that connects the connecting rod to the crankshaft) performs a rotary motion. Assume that the mass of the connecting rod is concentrated at two points, one of which is reciprocating with the piston mass and the other rotating with the crankshaft. The following relationship must therefore apply:

$$m_{connectingrod} = m_1 + m_2 m_1 b = m2a$$

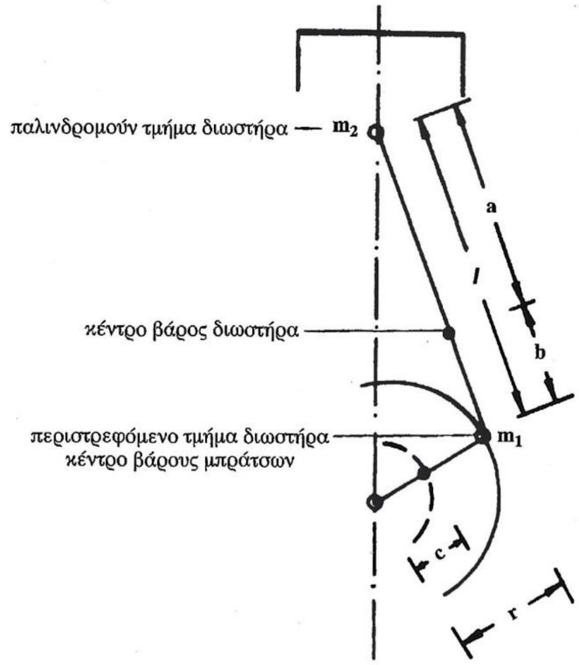


Figure (4.1.6) Reciprocating and rotating section connecting rod.

From these equations it follows that:

$$m_1 = m_{connectingrod}\frac{a}{1}$$

And

$$m_2 = m_{connectingrod} - m_1$$

The mass m1 is the rotating part of the mass of the connecting rod. The centres of gravity of the rotating masses of an ICE piston have different distances from the centre of rotation. For this reason, the centre of gravity of all these rotating masses is concentrated at a distance equal to the radius of the crank r. Thus, the real mass is replaced by a substitute mass which must produce the same result as the real mass. The equation of the substitution mass ratio of all masses is mrepl=mreal/r, where mrepl, is the substitution mass, mreal is the real mass, c is the distance of the real mass from the centre of rotation and r is the distance of the substitution mass from the centre of rotation. The mass of the rotating part of the connecting rod, like the mass of the hub, is concentrated at a point at a distance r from the crankshaft and therefore does not retreat. The masses of the arms of the crankshaft are withdrawn. Mrepl.arm=marmc/r. The relationship giving the forces of the rotating masses of the reciprocating engine is:

$$F_r = (m_{hub} + m_1 + 2\mathrm{m}_{arm}\frac{c}{r})r\omega \qquad (4.1.10)$$

mhub is the mass of the crankshaft hub, m1 is the mass of the rotating part of the connecting rod, 2marm, the mass of the crankshaft arms.

The reciprocating masses consist of the total piston mass (piston, piston springs, piston pin) and the reciprocating part of the connecting rod mass m2. The reciprocating forces are calculated by means of the relationship:

$$F_{recipr} = (m_{totalplunger} + m_2)r\omega^2(\cos(a) + \lambda cos(2\mathrm{a})) \qquad (4.1.11)$$

$m_{totalplunger}$ is the mass of the piston and m_2 is the reciprocating mass part of the connecting rod.

The piston is subjected to a force F, which is the vector sum of the gas and mass forces. This force can be broken down into a force perpendicular to the direction of application K and a force in the direction of the connecting rod S (Figure(4.1.7)):

$$K = Ftanh(b) \text{and } \Sigma = F/\cos(b) \quad (4.1.12)$$

The force K (vertical force) is taken up by the walls of the cylinder. The force in the direction of the connecting rod is transferred through the connecting rod to the hub of the crankshaft and is broken down into a radial force P and a tangential force T:

$$P = \Sigma cos(a+b) \text{and } T = \Sigma sin(a+b) \quad (4.1.13)$$

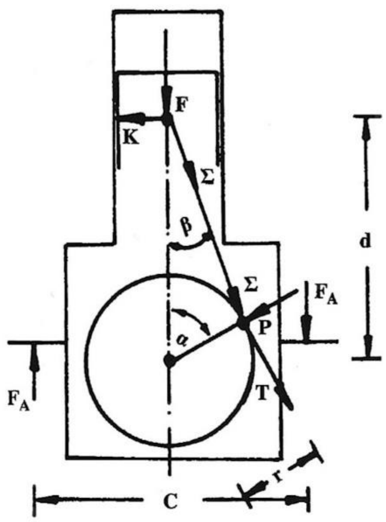


Figure (4.1.7) Forces of the reciprocating engine.

The radial force and the tangential force act on the crankshaft and the crankshaft support bearings. Due to the tangential force T we have the torque:

$$m_{torsion} = T_r \quad (4.1.14)$$

Angle β is negative when moving from the ADP to the LDP.

The piston of a rotary piston engine consists of three surfaces. Thus, we have three gas forces at the same time. These gas forces can be broken down into tangential and radial forces. Therefore, we have three tangential and three radial forces. All these forces (six) act on the support points of the camshaft. The torque of the camshaft is due to these forces. Mass forces are created on the piston and the camshaft. These are purely centrifugal forces. The direction of influence on these forces passes through the center of gravity of the camshaft so that it does not cause torque. The two gears necessary to achieve helical motion must carry only frictional forces and during the change in the number of revolutions of the piston also forces due to acceleration.

4.2 Internal Combustion Engine Torque Diagram

The torque diagram of an ICE piston-piston palladium ICE is developed below. In the torque diagram, the tangential force is studied in relation to the crank angle or the stroke of the crank hub impeller. The tangential force can be calculated by the relationship (4.1.13). It is, however, easier to determine by a graphical procedure. This method is developed below. Since the torsional force is directly related to the mass and gas forces, these must first be analyzed. The gas forces are obtained with the aid of the illustrative diagram. The indicative diagram is converted into a gas force path diagram by a suitable change of data reference. The gas forces are the result of the overpressure of the gases on the piston surface. If the absolute pressure is represented in the indicative diagram, the atmospheric pressure at the axis of the end position must also be considered. In the gas force path diagram, the reciprocating mass force is shown with a negative sign. The reciprocating mass forces shall be taken with opposite sign, so that the recommended ones can be obtained directly from the gas and mass force values as the distance difference of the gas and mass force diagrams. The crank circle is then drawn at a distance equal to the length of the connecting rod next to the force path diagram (Fig. (4.2.1)). the distance difference between the gas and mass force diagrams. Then draw the crank circle at a distance equal to the length of the pushrod next to the force-path diagram (Fig. (4.2.1).

The tangential force for each crank angle is calculated by the following procedure. The desired crank angle value in the crank circle is determined. Adjust the position of the piston for the selected crank angle. The intersection of the crank circle-angle intersection to which the crank angle belongs shall be used as one end of the pusher, and the other end of the pusher shall intersect the axis of the force-displacement diagram distances at the corresponding position.
At the point of intersection, the component forces of the masses and gases are shown. The difference in distance between the two curves is given as a vector, whose starting point is on the curve of the mass forces. This vector is also inscribed in the crank circle. When the direction of the vector points upwards on the path force diagram, it is plotted on the crank circle towards the inside of the circle. While when it points downwards, then we plot it towards the outside of the crank circle. In the crank circle figure, T1 is carried vertically, starting from the end of the F1 vector, until it intersects the extension of the connecting rod. This is the requested tangential force (torsional force). This vertical is given as a vector whose edge is always directed towards the extension of the connecting rod. If the vector points upwards, then we have a positive direction of motion (the torsional force acts in the direction of motion). The torsional force is plotted on the torque diagram, above the corresponding crank angle. If we take the actual vector lengths from the shape of the crank circle, then the measure of comparison of the torsional force diagram is the same as that of the force-displacement diagram. The above graphical diagram shows the process of calculating the torsional force for a cylinder. For multi-cylinder engines we follow the same procedure. Since the diagrams of all cylinders are the same, with the only difference being the phase difference (in crank angle) all diagrams are drawn first and then transferred to the general force-force diagram of the engine.

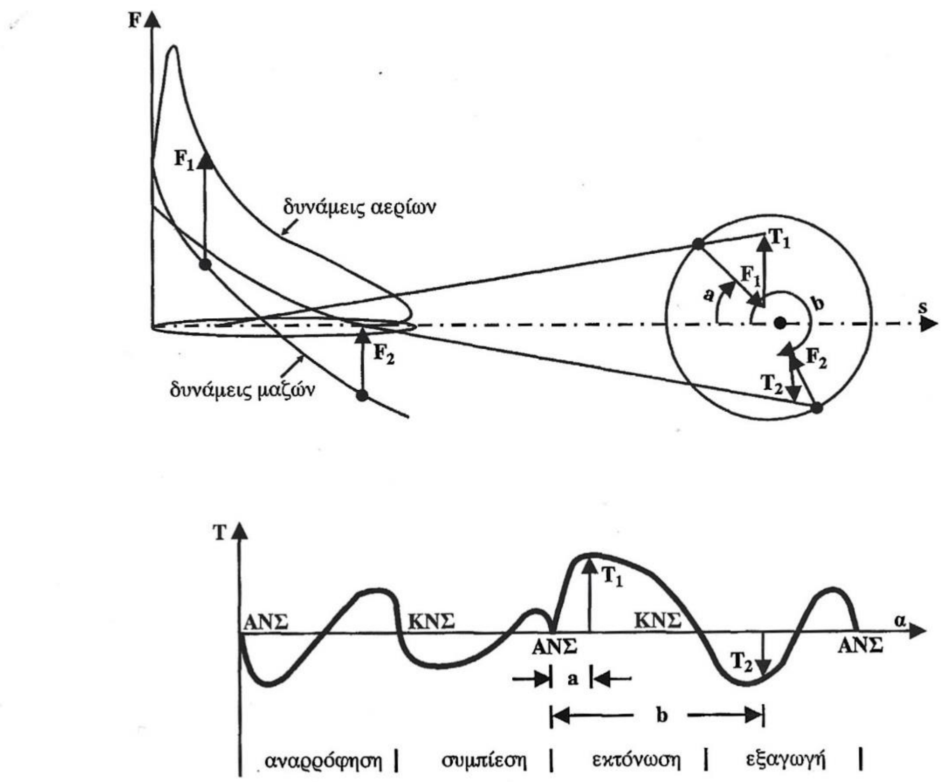


Figure (4.2.1) Graphical calculation of torsional force of reciprocating ICE.

4.3 Mass Force and Torque Balance

Mass forces create oscillations because of their variable periodicity. These oscillations are transmitted to the mounting points and the engine environment in general. For this reason, great care is taken to balance the mass. In multi-cylinder engines, in addition to mass forces, mass moments are also present. Thus, when we refer to mass balancing we mean using crankshaft sections and placing counterweights on the crankshaft to minimize the forces and moments of the masses.

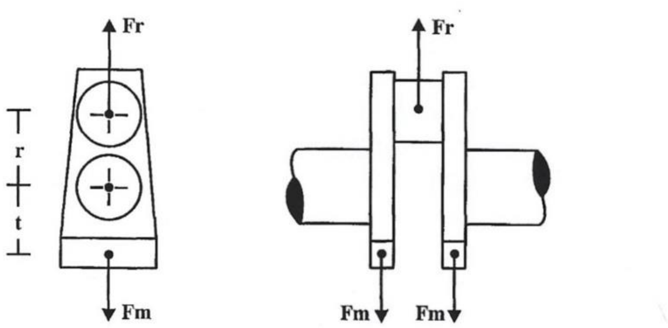


Figure (4.3.1) Counterweight on the crankshaft.

In the single-cylinder engine, only the rotational and reciprocating mass forces are shown, not the mass moments. The rotational mass force (Fig. (4.3.1)) is balanced by using two counterweights on the crankshaft.

The size of the counterweights is calculated using the equation:

$$F_r = m_r r\omega^2 = 2\mathrm{m}_{counterweight} t\omega^2, m_{counterweight} = m_r \frac{r}{2\mathrm{t}} \qquad (4.3.1)$$

Mcantiweight is the mass of the counterweight to balance the force of the rotating mass, mr is the rotating mass, r is the radius of the crankshaft and t is the distance from the centre of gravity of the counterweight to the centre of gravity of the crankshaft. The force of the reciprocating mass is calculated by means of the relationship Frecipr = mrecipro rω2(cosa + lcos2a). This mass force shall be broken down into first and second order first and second order
mass forces.

$$F_1 = m_{recipro} r\omega^2 cosa \qquad (4.3.2)$$

And:

$$F_2 = m_{recipro} r\omega^2 \cos(2\mathrm{a}) \qquad (4.3.3)$$

he first-order mass force varies with the rotational speed of the crankshaft by cos(a). The second-order mass force, in contrast, depends on twice the crankshaft angle α(cos2a). Therefore, by using counterweights on the crankshaft, only the first-order mass force is balanced. Since the reciprocating mass force is in the direction of the vertical axis of the cylinder and the counterweights move with the crankshaft, only the vertical component of the counterweight force is necessary for balancing. The horizontal component is an unwanted additional force (Fig. (4.3.2)).

To keep the horizontal component from being too large, 50% of the force of the first-order masses is balanced by counterweights. The relationship for calculating the counterweights is given by the equation, zF1=ζmreciprorω2 cosa=2mcounterweighttω^2:

$$m_{counterweight} = \zeta m_{recipro} \frac{r}{2\mathrm{t}} \qquad (4.3.4)$$

mcountertweightis the mass of the counterweight to balance the first-order mass forces and z is the percentage of the first-order force that is balanced (usually τ = 0,5). In single-cylinder engines, the second-order mass force is also balanced. To achieve this, high manufacturing costs are required. Figure (4.3.3) shows an engine in which the rotational and reciprocating forces of the first- and second-order masses are balanced.

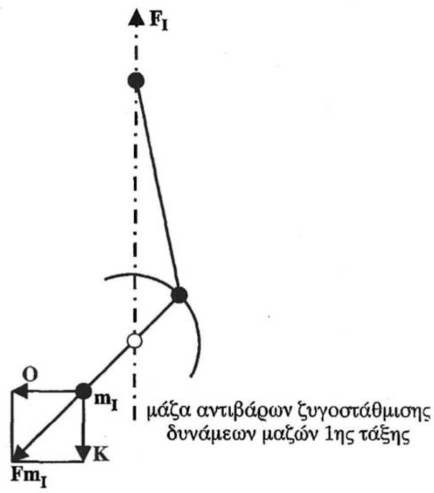


Figure (4.3.2) Analysis of counterweight force in horizontal and vertical component.

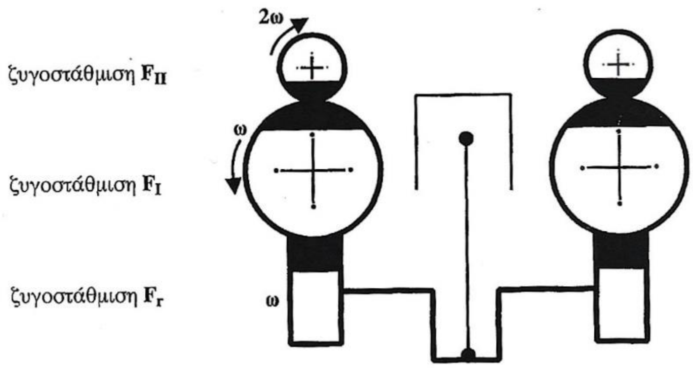


Figure (4.3.3) Complete balancing of a cylinder engine.

The counterweight axes are in opposite directions, so that the horizontal components of the force oppose and balance each other. The external counterweights of the second-order mass force move at twice the speed of the crankshaft.

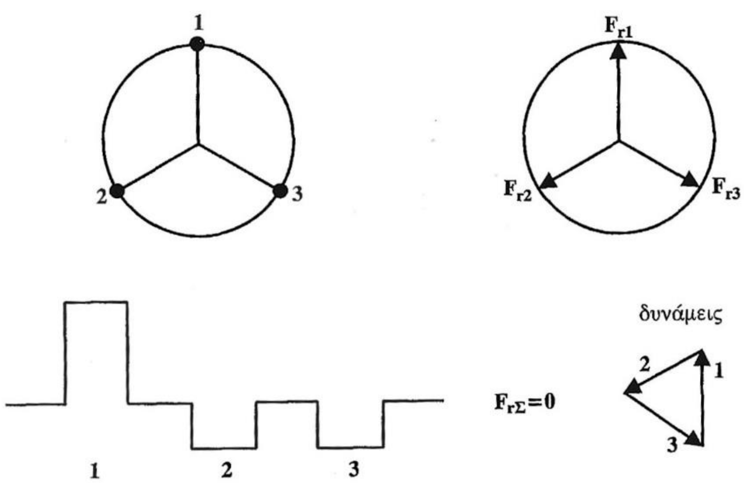


Figure (4.3.4) Graphical calculation of 3-cylinder crankshaft.

To achieve a balance of mass and torque forces in in-line engines, the mass forces occurring in each cylinder shall be calculated from the formulae in this chapter and added to obtain the recommended force. Balancing is successful if the resulting force is zero or as small as possible. Because the forces do not act on the centre of mass of the engine, additional mass moments occur, which are added to the resultant. The magnitude of all these elements depends largely on the type of crankshaft. For this reason, the crankshaft must be designed so that the individual forces cancel each other out. To solve such problems, the recommended force for the various types of crankshaft must be determined, which is most quickly achieved by the graphical method. First, the cross-section along and across the crankshaft is drawn (Fig. (4.3.4)).

The cross-section to the left and above the longitudinal cross-section is called, because of its appearance, the star crank arrangement. The crankshaft numbering follows the longitudinal cross-section of the crankshaft and is carried over to the star crankshaft arrangement. The rotating mass forces of each cylinder are calculated by relation (4.1.6). For vector addition, all forces, after setting an appropriate scale, are transferred to a parallel uniform cross-sectional plane, (Fig. 4.3.4). top right. The above positional geometry of the generated forces has the same form as the star configuration of the crank. In the force diagram, the forces to find the resulting FrS are added and placed at the correct position in the force diagram. On one rotation of the crankshaft, the element rotates with it by a constant amount, so for another crankshaft position the element does not need to be repositioned, but only needs to be rotated according to the angle of the crankshaft. In in-line engines with an even firing order and with more than two cylinders, the component is zero.

The first-order mass forces are calculated by means of relation (4.3.2). Their maximum value is F1m=mreciprorω2 .The instantaneous value is obtained graphically if the maximum value is plotted in the direction of the crank arm in the crank star arrangement and projected on the cylinder axis. To determine the component by this procedure, all the instantaneous values of the first-order mass forces must be formed and then added vectorially. This procedure, however, is time consuming. The same component is more easily obtained if the maximum values are plotted in the direction of the crankshaft arm and these forces are added to the force diagram, (Fig. (4.3.5).

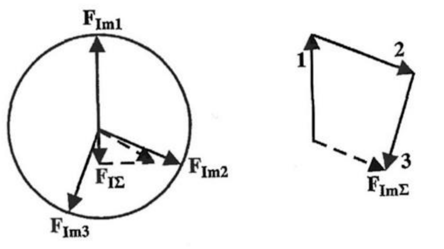


Figure (4.3.5) Graphical finding of the component of the 1st order mass forces FIS in a 3-cylinder engine of uneven ignition sequence.

The component of the maximum Flm.S values is projected on the axis of the cylinder. This projection is the desired component of the 1st order mass forces FIS .

Since the force diagrams of the rotational mass forces and the 1st order mass forces are exactly the same except for the length of the vector, it is clear that when the component of the mass forces is zero, then the component of the 1st order mass forces is zero. By rotating the crankshaft, the component of maximum values rotates simultaneously together. For the new position of the crankshaft, the maximum value component need only be projected onto the cylinder axis to obtain the 1st order component of the mass forces.

The 2nd order mass force is obtained by means of relation (4.3.3). Its value varies at twice the angle of the crankshaft, but it acts like the first-order mass force only on the cylinder axis. To determine the components, the maximum values of the forces of each cylinder are plotted, like the first-order mass force, in the force arrangement diagram, (Fig. (4.3.6):

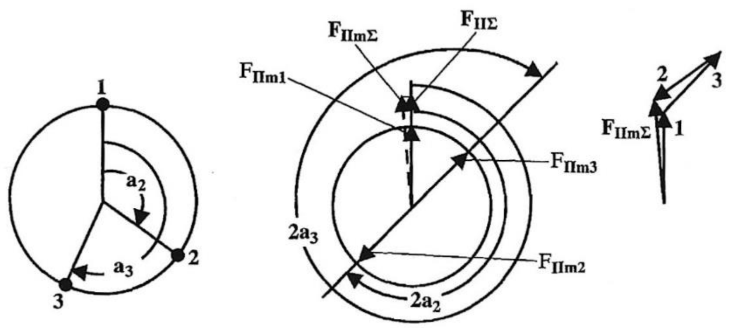


Figure (4.3.6) Graphical calculation of the 2nd order mass component of the FIS mass forces for a 3-cylinder engine with non-uniform ignition sequence.

These forces do not have the same direction as the radii of the crank arrangement, due to (cos(2a), but must be plotted on the force diagram at twice the angle. In the force layout diagram, these forces are synthesised into a component of the maximum FIImS values and transferred to the position layout diagram. Their projection on the axis of the cylinder gives the required component of the second order mass forces FIIS. In one rotation of the crankshaft by an angle α, the maximum value component in the position diagram is rotated by an angle 2a. The required coordinate is then its projection on the cylinder axis.

Because the mass forces are at some distance from the centre of gravity of the engine, mass moments are developed. These moments are calculated by the product of the force times the distance from the centre of gravity. To determine them accurately, the position of the engine's centre of gravity must be determined. Usually this time-consuming procedure is avoided and the engine centre of gravity is assumed to be at the centre of the crankshaft. The error in the calculations is considered to be negligible. Thus, the reference point for the mass moment calculations is the centre of the crankshaft in the longitudinal section. Like the force, the torque can be.
be represented as a vector. The magnitude of the torque is shown along the length of the vector. The vector is applied perpendicular to the plane of action of the torque and has a direction balanced by clockwise rotation in the direction of the torque. Because a torque vector allows torque to be transferred parallel to its plane of action, all vectors passing through the center of the crankshaft are drawn in the plane of action perpendicular to it. The resulting layout diagram may also be drawn using the star-

shaped crankshaft layout. In addition, however, all torque vectors, as well as the crankshaft arms, shall be drawn so that they rotate 90 degrees parallel to the direction of rotation. The direction of the force vectors to the left of the reference point shall be directed outwards from the centre of the star crankcase assembly, while the force vectors to the right of the reference point shall be directed towards the centre of the star crankcase assembly.

The mass moment component is easier to calculate if the individual vectors are not plotted with a 90° rotation parallel to the motion, but are plotted according to the crank arm arrangement. Then, after these vectors make up the component force in the MrS. torque layout diagram, the torque vector is driven to its correct position by symmetrical 90° rotation (Fig. (4.3.7)).

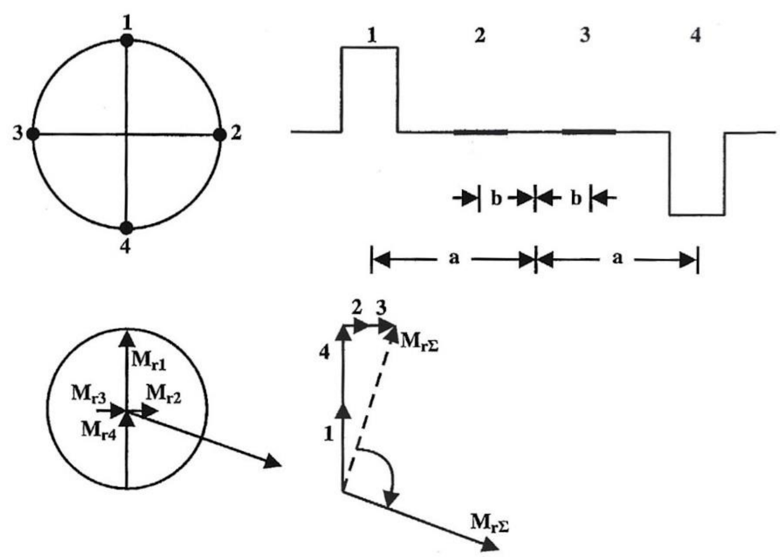


Figure (4.3.7) Graphical calculation of the torque component of the rotating masses MrS for a 4-cylinder engine.

The reciprocating mass forces act only along the axis of the cylinder. For this reason, all moment vectors are applied perpendicular to the plane of the cylinder axes. For the determination of the moment component, the maximum values shall be taken as a basis, just as the component of the 1st order mass forces is determined. First, the maximum torque values for all cylinders shall be calculated and plotted on the positioning diagram, (Fig. (4.3.8). After the MlmS component is formed in the torque layout diagram, it is transferred to the position layout diagram and projected on the axis of the cylinders .By projecting 90° (in the direction of the same rotation), the actual torque component of the 1st order MIΣ forces is obtained.

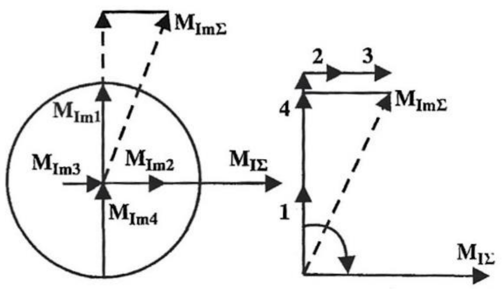


Figure (4.3.8) Graphical calculation of the moment component of the 1st order Mis masses for the crankshaft Fig.

The 2nd order mass moment vectors appear perpendicular to the plane of the cylinder axes and their magnitude varies with twice the crank angle. After calculating the maximum values of the second order mass moments, their vectors are plotted with doublings for a crankshaft in the position vector diagram (Figure (4.3.9)):

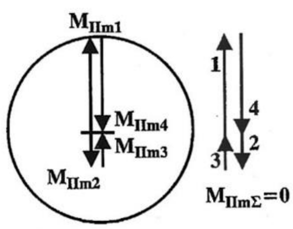


Figure (4.3.9) Graphical calculation of the component of the moments of the second-order Mies masses for the crankshaft in figure (4.3.7).

The component of the MImS moment ordering diagram is transferred to the position ordering diagram and projected onto the cylinder axes. The actual vector of the 2nd order MIIS mass moment component is obtained by rotating the projection 90° parallel to the direction of motion.

CHAPTER 5

5 Basic Reciprocating Part of Internal Combustion Engines

5.1 Basic Reciprocating Part of Internal Combustion Engines

The cylinder diameter D, the piston stroke H and the number of cylinders Z are described as the main dimensions of a reciprocating engine, because they define the basic dimensions of an engine. The basic dimensions of an engine must be determined prior to its manufacture. To calculate them, the manufacturer considers the following elements: net power and total cubic capacity if the engine is subject to cubic capacity taxation (cars, motorcycles), net power and number of revolutions for all other engines. In addition to the above information, the type of engine should be known. The useful power, the buyer's desire and the manufacturer's delivery time all play a role in the choice of engine type. The following classification helps in selecting the type of construction of an engine.

The useful power up to 7 kW, two-stroke Otto engines, four-stroke Diesel engines, four-stroke engines are excluded.

Motorcycle engines, two-stroke and four-stroke Otto engines.

Passenger car engines, four-stroke Otto engines, four-stroke Diesel engines, two-stroke engines excluded, Otto engines.

Truck engines, four-stroke diesel engines, for small trucks and four-stroke Otto engines.

Marine engines, plant engines, up to 13,000 kW, two-stroke and four-stroke diesel engines, above 13,000 kW, two-stroke diesel engines. The largest engines in operation today are 12-cylinder engines with 36 000 kW.

A few observations will help to better understand the above classification. Two-stroke engines are economical due to their simple construction but have very high specific fuel consumption. For this reason, they are also suitable for low power in non-continuous applications (mopeds, etc.). Four-stroke engines are more expensive to build but more economical to operate. The smaller

diesel engines have specific fuel consumption. However, due to heavier construction and the addition of the injection system, their manufacturing price is high. Delivering the same power at the same rpm, the size and weight of two-stroke diesel engines are smaller than four-stroke diesel engines. For this reason, two-stroke diesel engines are preferred in the manufacture of large engines.

For the calculation of the main dimensions of cars and motorcycles with based on total displacement, the relationship is as follows:

$$D, H, V_h = \frac{\pi D^2}{4} H z$$

With the help of the embolism ratio:

$$\kappa = \frac{H}{D}$$

arising:

$$V_h = \frac{\pi D^3}{4}\kappa z$$

and from this:

$$D = \left(\frac{4V_h}{z\pi\kappa}\right), H = \kappa D \qquad (5.1.1)$$

In relation (5.1.1), values derived from experience are presented, while it is practical to check the calculations with the help of the average piston speed:

$$u_{mid} = 2\text{Hn} = u_{allowable}$$

the permissible, the permissible average piston speed. The necessary number of revolutions used in this relationship is derived from the appropriate conversion of the power formula:

$$n = \frac{P_\varepsilon}{\rho_\varepsilon V_h z i}$$

If the calculated average piston speed is close to the permissible speed, then the main engine dimensions have been correctly determined. Otherwise, the calculations should be repeated with improved empirical values.

Empirical values for motorcycle engine cylinder numbers z = 1, 2, 3, 4 and for passenger car engine z = 4, 6 or 8 (V-type construction). Many cylinders give uniform torque and good mass balance, but the engine becomes more expensive.

Empirical values for the piston ratio k = 0,6 to 1,1. A small κ is preferred because it reduces the ramming and thus the average speed of the piston. Shorter stroke means reduced build height and low average piston speed and ultimately long engine life. The diameter of the cylinder with small and becomes large. Consequently, large valves can be built, resulting in good cylinder filling and high average effective piston pressure. The disadvantages of the small stroke ratio are the larger piston diameter with correspondingly large forces on the piston and bearings. In the case of the high compression ratio, the combustion chamber is too flat resulting in poor combustion and greater heat loss through the walls.

For the average actual piston pressure of a motorcycle engine, for a four-stroke Otto engine, with re= 8.5 to 10 bar, while for a two-stroke engine pε= 6 to 7.5 bar. Passenger car engine, four-stroke Otto engine, pe = 7 to 10 bars. Passenger car engine, four-stroke diesel engine, PE = 5,5 to 6,5 bars.

The average speed of the piston must be chosen as low as possible to avoid excessive wear on the piston springs, piston, and cylinder liners. average allowable0 u = 10 to 17 m/s.

The basis for calculations of the main dimensions of engines, based on the net power, starts with the net power formula:

$$P_\varepsilon = p_\varepsilon V_h zni$$

With:

$$V_h = \frac{\pi D^3}{4\kappa}$$

We have:

$$P_\varepsilon = p_\varepsilon \frac{\pi D^3}{4} \kappa zni, D = (4 \frac{P_\varepsilon}{p_\varepsilon \pi \kappa zni}), H = \kappa_D \qquad (5.1.2)$$

Based on the control with the permissible speed relation, empirical values for pε, k, Z are used in relation (5.1.2).

Empirical values for the average actual piston pressure, diesel truck engine (four-stroke), no supercharging, pe = 6 to 9 bar, supercharging, pe = 9 to 11 bar.

Medium-speed four-stroke diesel engines (n = 500 rpm), without supercharging, pe = 5 to 7 bar, supercharged pe = 8 to 10 bar. With supercharging and air-cooled charging pe = 12 to 20 bar.

Low-speed (n = 500 rpm) two-stroke diesel engines with supercharging and air cooling, with pE = 9 to 15 bar.
In case of supercharging, a compressor supplies the fresh air at overpressure, so that the degree of supercharging becomes λ1 > 1. At a high degree of supercharging, the air temperature in the supercharging area increases due to the strong compression. Therefore, the air is cooled before it enters the engine to reach its original temperature.

As far as the compression ratio is concerned, the diesel engines of cars and trucks k = 0,9 to 1,2. Medium-speed four-stroke diesel engines, k = 1,2 to 1,4, low-speed two-stroke diesel engines, k = 1,8 to 2,2

As regards the number of cylinders of diesel engines of trucks 2 = 6 or 8. For 8 cylinders the V-type construction is chosen.

Stationary diesel engines and marine diesel engines, in-line type construction, z = 1 to 12, V-type construction z = 8 to 20. In a V-type engine the cylinders are in a V-shape arranged in two rows and for a large number of cylinders the engine and especially the crankshaft is smaller and less flexible.

As for the allowable average piston speed, diesel engines for trucks uallowable = 10 to 11 m/sec; medium-speed, four-stroke diesel engines, uallowable = 6 to 8 m/sec; low-speed, two-stroke diesel engines, uallowable = 6 to 7 m/sec.

5.2 Piston

The piston operates under very difficult conditions. It is subject to very strong mechanical and thermal stresses. Figure (5.2.1) shows a piston with the main cooperating parts.

Under these conditions it is required to perform the following functions, the conversion of pressure energy into mechanical work. The sealing of the cylinder space from the crankcase. The linear movement of the upper end of the pusher in rod engines. The adjustment of the opening of the intake and exhaust ports in two-stroke engines. In order for the piston to meet these requirements, it must have the following characteristics: low mass, so that the reciprocating forces of the masses remain small, even at high operating speeds high rigidity at the top of the piston, flexibility at the shaft and elasticity at the body high strength in the spring area, even in the event of possible breakage high heat resistance good thermal conductivity so that large temperature changes do not occur in the piston material small thermal expansion so that there are small operating clearances.

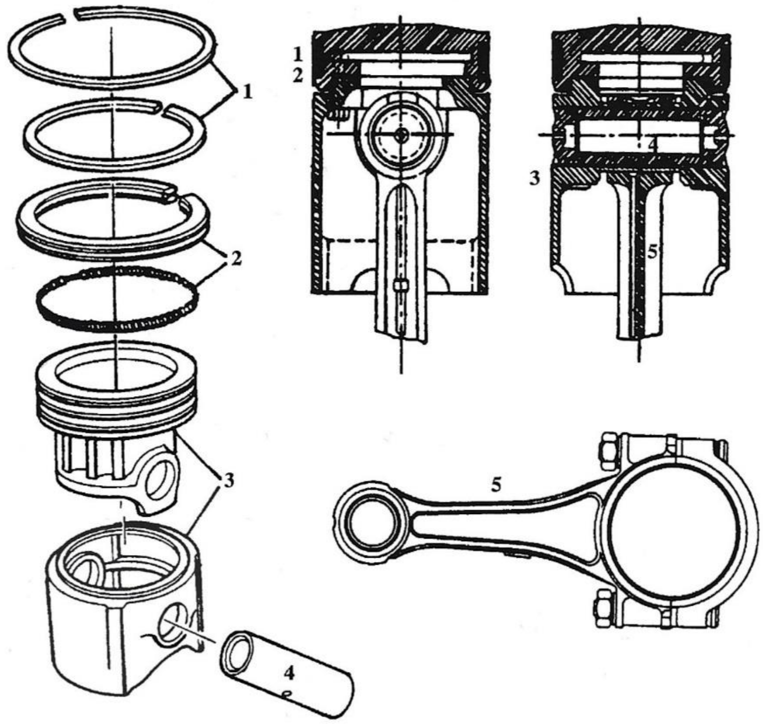


Figure (5.2.1) 1 compression springs, 2 oil springs, 3 piston, 4 piston pin, 5 connecting rod.

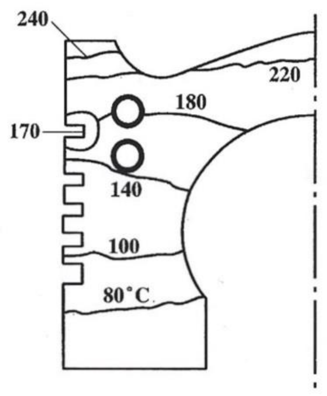


Figure (5.2.2) Piston temperature under full operation.

The following ignition pressures are applied to the piston head: 40-70 bar in the Otto engine, 60-100 bar in the Diesel engine without supercharging, 60-140 bar in the Diesel engine with supercharging.

The piston temperature is influenced by many factors, such as engine operating mode, combustion and cooling mode, engine load, etc. The piston temperatures developed in a four-stroke Otto engine under full operation are shown in Figure (5.2.2).

Fig. (5.2.3) shows the major components of the piston. D is the piston diameter, D1 is the inner diameter, D2 is the length of the free pin, L is the length of the piston,L1 is the length of the rod, L2 is the compression height. d is the diameter of the pin, b is the thickness of the head, e is the height of the second spring, f is the height of the first spring and c is the firing height.

The position of the piston pin is determined by two factors. To prevent the piston from tipping over, the center of gravity of the piston must be located on the pin axis. Otherwise, the pin must be firmly fixed in the center of the piston rod to transfer the vertical force evenly to the cylinder walls. Since it is not possible to meet both conditions simultaneously, the pin is positioned slightly above the center of the piston rod and to reduce noise the pin position is shifted from the vertical axis. (Fig. (5.2.4) The center of the piston pin shall be shifted by 1 to 2 mm towards the pressure side of the piston. By shifting the pin, the change of the piston position in the cylinder takes place just before the and the overturning motion is not so strong.

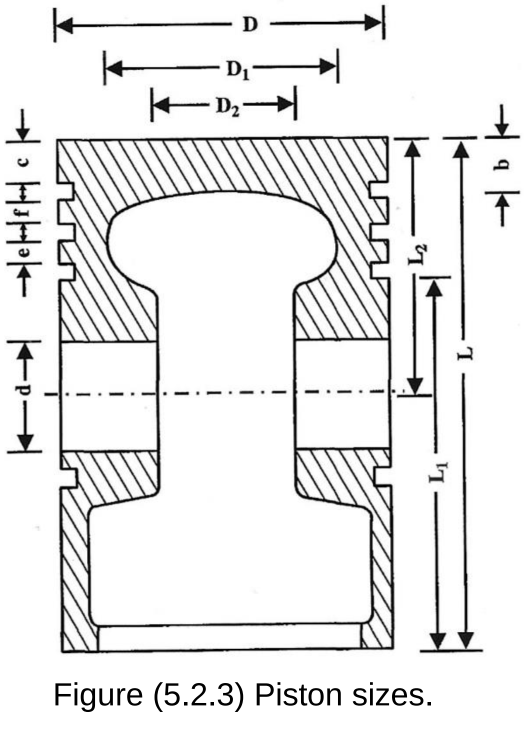


Figure (5.2.3) Piston sizes.

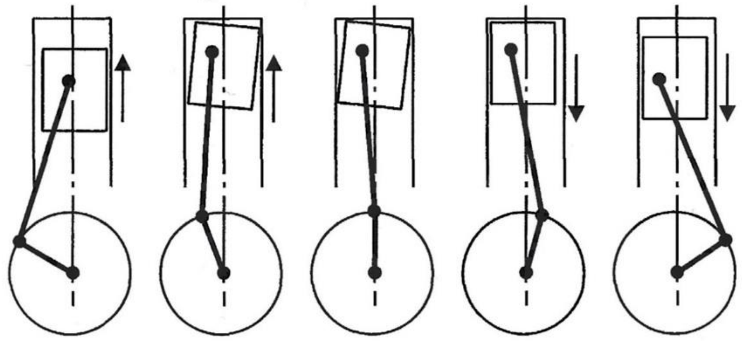


Figure (5.2.4) Change of piston position.
(Piston displaced towards pressure side).

Small and medium-sized pistons (up to about 500 mm in diameter) are mostly made of cast aluminium alloy. Very large pistons in high demand are cast from a special aluminium alloy. Large pistons consist of two or more pieces. The head of the piston (steel or cast iron) is attached to the body (cast iron or aluminium alloy) with screws. Various aluminium alloys are used to make the piston. In addition to aluminium, the alloys contain 11-25 % silicon, and, as appropriate, 1-2 % copper, nickel, and magnesium, and less than 1 % iron, titanium, manganese, and zinc. The piston during operation expands more than the cylinder enclosing it and for this reason a relatively large gap (in cold condition) must be provided when assembling the engine. The external shape of the piston in the cold state is rounded oval, so that when the engine is operated under the influence of temperatures and masses it assumes its cylindrical shape. Keeping the piston expansion at low levels benefits in low operating noise, low oil consumption and low friction.

5.3 Piston springs

Piston springs are divided according to their function into two categories, compression springs and oil sealing springs, (Fig. (5.2.1). The main purpose of compression springs is to seal the combustion chamber, but they also influence the amount of oil remaining on the cylinder walls. The main function of the oil sealing springs is to direct the remaining oil to the crankcase. Piston and spring lubricating oil is sprayed from the crankcase position onto the cylinder walls, and in large engines it is guided through special holes in the cylinder. But the springs also serve to induce heat from the piston to the cylinder walls. The maximum amount of heat is removed by the upper compression spring, which, because of its position, is the least lubricated and ' this is what suffers the most wear. The heat treatment of hardening and chrome plating largely avoids mechanical and corrosive damage.

The wide variety in the form of springs is necessary because of their wide range of use. Thin springs are quickly adapted to the shape of the cylinder because their contact surface is initially very small. Trapezoidal springs are used where there are residual quantities of lubricating oil and fuel. The springs, because they are mounted in their groove, especially the trapezoidal ones, help to remove the dirt. The piston springs must constantly press against the cylinder walls to achieve a good seal. Increased pressure tension is necessary to prevent the so-called "play" of the springs. In two-stroke engines, the pressure must be lower so that the springs do not get stuck in the grooves and become damaged.

The material used for the springs is special cast iron and a special casting method is used. They are then formed into a non-circular shape and cut. In this way, symmetry of the distribution of the radial pressure of the springs is achieved when they are assembled in the cylinder.

5.4 Piston Crown

The piston pin transfers the forces between the piston and the connecting rod. It is manufactured from steel after suitable hardening and surface grinding. Due to the forces exerted, the piston pin is bent, deformed into an oval shape, and subjected to shear stress. For the calculation of its dimensions, the permissible stresses are not given, but the deformations at the piston hub are given, because this is where the damage that leads to piston destruction starts. If the deformations of the pin can be kept low, then the stresses developed remain within the permissible limits.

5.5 Drivers

The piston (fig. (5.2.1)) connects the piston to the crankshaft. It is made of heat-treated steel with a hardness of 600-700 N/mm2. The cross-section of the stem is circular or of the double-T type. For lubrication of the pin or even for its cooling, oil is led through a hole along the stem of the connecting rod. The length of the connecting rod is chosen as short as possible to reduce the height of the engine, which results in a lower weight. The minimum limit for the thrust rod construction ratio is λ = 1/3,4. The upper end of the connecting rod is connected to the piston pin bearing and is made from one piece. The lower end of the connecting rod shall be split for manufacturing (assembly) purposes. Because the connecting rod must pass through the cylinder during assembly, it is necessary, with a full crank pin, that the lower part of the connecting rod be split. The piston rod is stressed due to compression and tensile forces. The stresses at the ends of the connecting rod are complex and comparative stresses are used in the calculation. The comparative stresses do not give the actual values of the stresses. The piston rod is calculated in compression and tension and, when its length is long, also in bending.

5.6 Crankshaft

On the crankshaft (figure (5.6.1)), the reciprocating motion is transformed into rotary motion.

Steel is used as a construction material. The small steel crankshafts are forged in molds, while the larger crankshafts are first free forged, and care is taken to rotate the shaft keys to the correct position and then the crankshaft is finished. The crankshafts of large engines (cylinder diameter of 500 mm or more) are assembled in sections. At this point, semi-finished shafts differ from solid shafts. Semi-fabricated shafts incorporate the shaft hubs into the crankshaft arms, which are one-piece (two arms and a crankshaft hub and then the crankshaft ends. The crankshafts of large engines (cylinder diameter of 500 mm or more) are assembled in sections. At this point, semi-assembled shafts differ from solid shafts. Semi-manufactured shafts incorporate the shaft hubs into the crankshaft arms, which are one-piece (two arms and one crankshaft hub. In contrast, solid crankshafts are assembled piece by piece. Specially treated steel with a hardness of 800-900 N/mm is used for the manufacture of crankshafts. The smaller shafts are hardened at the bearing points.

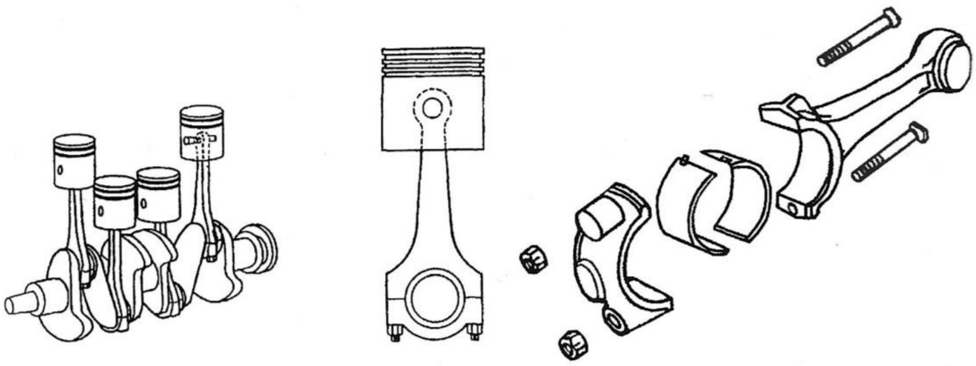


Figure (5.6.1) Crankshaft.

It is not possible to accurately calculate the strength of the crankshaft. The stress on the shaft is so complex, the changing forces of the masses and gases, the torsional and flexural oscillations and the active forces are so complex in form and magnitude that only simplified calculations are possible.

Crankshafts are vibration-resistant structures consisting of elastically coupled masses. If a periodically varying force acts on such a mechanical system, an externally excited or forced oscillation takes place. If the frequency of excitation of the force becomes equal to the natural frequency of the crankshaft, resonance occurs, and the amplitude of the oscillation becomes very large. The additional stress due to the oscillations must not exceed the strength of the material, otherwise the shaft will break.

Three types of oscillations can occur in the crankshaft: longitudinal oscillations: the shaft oscillates along the longitudinal axis; flexural oscillations: the shaft oscillates perpendicular to the longitudinal axis; torsional oscillations: the shaft oscillates around the longitudinal axis. Torsional oscillations are the most dangerous because they are the most common cause of crankshaft failure.
To avoid torsional oscillations, even if the shaft does not fail because undesirable situations occur, such as disturbance of the mass balance, excessive wear of the gears, noise, the calculation for oscillations must be carried out before the crankshaft is built. The following problems must be clarified: the type of natural oscillation and the natural frequency of the crankshaft, the resonance position and critical speeds, the amplitude of the oscillations and torsional stresses, impermissible large torsional oscillations.

5.7 Flywheel

The torque of a reciprocating engine is not constant but varies continuously during operation. However, motor-driven mechanisms generally require constant torque. Normalization of the torque curve is achieved by the addition of a flywheel, (Fig. (5.7.1). The flywheel saves energy and, as far as possible, keeps the operating speeds constant. If the motor torque is higher than the average torque, the operating speed is increased, and the flywheel saves energy. When the motor torque again becomes less than the average torque value, the operating speed is reduced, and the flywheel saves energy. For the flywheel to function as an energy saver, it must be possible to change the operating speed. The range of variations is an empirical value and depends on the motor providing the drive. Since at low operating speeds the variations are smaller, the empirical value is given in proportion to the variation of the average speed. This relationship characterizes the degree of uniformity (d). The angular speed is most often used to determine the degree of uniformity instead of the number of revolutions as ω:

$$\delta = \frac{\omega_1 - \omega_2}{\omega_\mu}$$

ω1 is the maximum angular velocity during an operating cycle,ω2 is the minimum angular velocity during an operating cycle and:

$$\omega_\mu = \frac{\omega_1 + \omega_2}{2}$$

the arithmetic mean. The value for the ohm is calculated from the number of engine operating speeds using the relationship:

$$\omega_\mu = 2\pi n$$

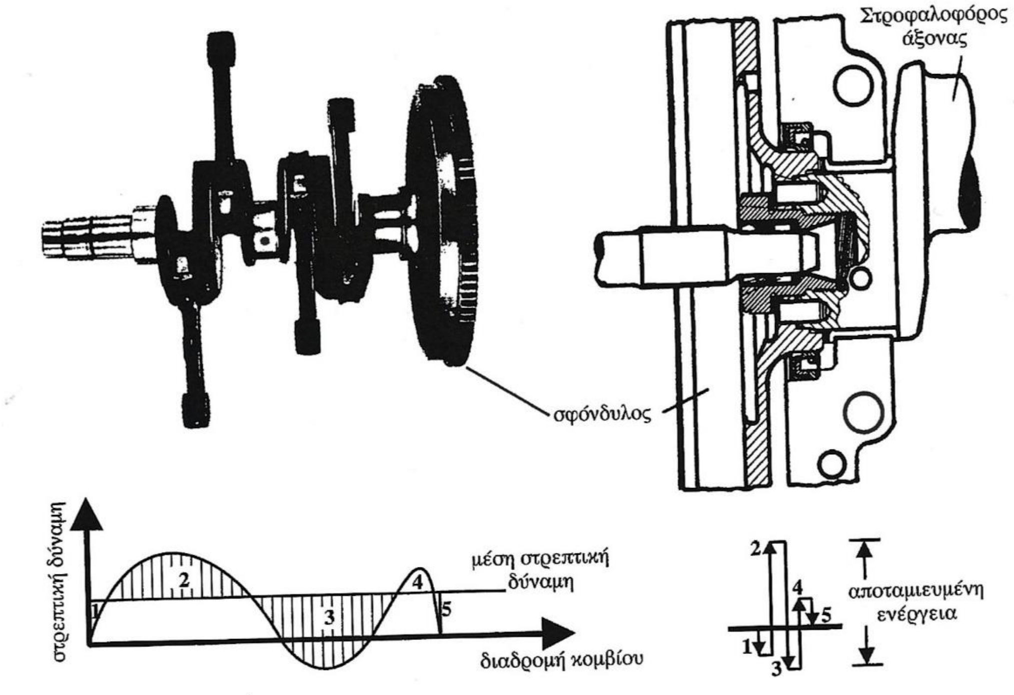


Figure (5.7.1) Flywheel and calculation of saved energy.

The following empirical values are used for the permissible degree of non-uniformity (d): power generator 1/300, vehicle engine 1/200, DC generator 1/150, textile machine 1/90, paper printing machine 1 /45, pumps and blowers 1/25.

The kinetic energy A stored in the flywheel is calculated from the relationship the relation:

$$A = \frac{J}{2}(\omega_1^2 - \omega_2^2)$$

J is the moment of inertia of the flywheel mass. This relationship is converted using the degree of non-uniformity to calculate the flywheel to:

$$A = J\frac{\omega_1\omega_2}{2}(\omega_1 - \omega_2)$$

and finally, it is done by adding the degree of non-uniformity:

$$A = J\omega_\mu\delta\omega_\mu$$

we have:

$$A = J\omega_\mu^2\delta$$

This relationship is used to calculate the moment of inertia of the flywheel mass. Usually, the moment of inertia of the crankshaft mass is not taken into account. A, which is the energy stored in the flywheel, is obtained by the following procedure. In the torsional force diagram, the torsional force is plotted against the travel distance. The area represents the work. The values of all surfaces above and below the horizontal average torsional force are taken and the values of these quantities are determined. A scale is then defined for the contained values of these surface areas and plotted as vectors. The vectors of the upper half of the average torsional force line plot shall point upwards. The distance of the outer edges of the vectors is the stored energy of the flywheel, (Fig. (5.7.1).

5.8 Cylinder

The cylinder surrounds and directs the movement of the piston. In small engines, the casting is made directly in the crankcase. This is a very economical solution, but in the event of cylinder failure, the engine must be completely disassembled to repair the damage. Engine repair is easier if a special liner is used. Wet or dry linings are manufactured. Wet liners cool well because they are directly surrounded by cooling water. Their walls are suitably constructed so that they can absorb the pressures of the gases. Dry liners have thin walls and are "pressed" into the crankcase. The heat dissipation in dry liners is not as good as in wet liners, The liners are made of good quality cast iron. The required performance characteristics of the liners are further improved by chrome plating, which increases resistance to mechanical as well as chemical damage. Since no lubricant is retained on the smooth chrome surface, porous chrome plating occurs. In air-cooled engines, an aluminium alloy cylinder is successfully used. Aluminium alloy absorbs three times more heat than cast iron and therefore the temperature distribution is more uniform and the intensity of heat transfer is lower. The working surface of these cylinders is also chrome-plated.

5.9 Valves – Timing Determination – Locks

Reciprocating internal combustion engines belong to the category of engines that run intermittently. A certain amount of the operating charge enters the cylinder, where it performs some work and is then expelled from the cylinder. This filling (filling) and emptying of the cylinder is called load cycling. The load-shifting behavior greatly affects the performance of the engine. Ideally, the load cycling should be such that all exhaust gases exit the cylinder and then the cylinder is refilled with fresh mixture. The method of load switching is different for two-stroke and four-stroke engines.

In the four-stroke engine, the load switching takes place during two strokes. The piston sucks the new charge through the intake valve and at the end of the operating time pushes the exhaust gases out of the cylinder through the exhaust valve. The combustion chamber, which is the space above the piston at the Top Dead Centre, can be emptied and refilled with the intake and exhaust valves simultaneously open, utilising the mass inertia of the working fluid. The opening and closing of the vanes is accomplished through the valves. The valves open and close relatively slowly because their initial and final speeds are zero. During intake and exhaust, there are large leakages and the operation of the valves at high speeds presents some problems due to the vibrations that can lead to breakage of the valve springs. The valve operation is shown in figure (5.9.1).

The push rod transfers power from the transmission cylinder to the rocker arm. The push rod is made of steel or aluminium alloy and its cross section is solid or hollow.

The rocker arm moves the valve and has lever arms of equal or different lengths. In arms of different lengths, the shorter lever arm should be on the side of the pushrod and then a shorter arm elevation allows for a longer valve travel. Small rocker arms are made of forged steel or steel plates. The valve clearance is adjusted on the rocker arm. The size of the valve gap should be selected so that when the engine is at operating temperature, there is still some "play" between the rocker arm and the valve so that the valve will close tightly and securely. A leaking valve will be burned by the high temperature exhaust gases escaping at high velocity, resulting in its destruction.

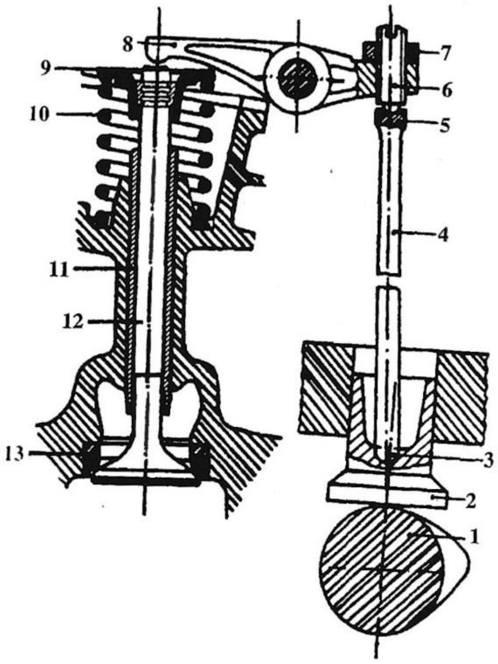


Figure (5.9.1) Valve operation, 1 camshaft, 2 drive gear, 3 ball, 4 push rod, 5 support bearing, 6 adjusting screw, 7 resistance nut, 8 rocker arm. Spring disc, 10 valve spring, 11 valve guide, 12 valve, 13 valve seat.

The valve spring must close the valve and keep it closed under vacuum conditions in the cylinder. All components of the transmission system must generally be in tension with each other (compressed). In most cases, helical compression springs are used. For low forces and low engine height, closed springs are manufactured. In high-speed engines, valve springs are often not used to prevent them from breaking due to vibration.

The valve is exposed to high mechanical and thermal stresses and to corrosion due to combustion products. At full operation, the temperature at the exhaust valve seat can reach 800 °C and at the intake valve 500 °C. Only high-strength steel alloys with high thermal resistance can withstand the stresses to which the exhaust valve is subjected. For a longer service life, the valve seat is shielded with a special high-strength chrome-nickel alloy. In addition, the temperature of the valve seat is reduced by 80 K when the valve is cooled with sodium. The hollow stem is 2/3 filled with sodium, which during operation liquefies, evaporates and helps to dissipate the heat from the seat to the stem and the valve drive system. The vacuum stem valve, among other things, is not as sensitive to deformation of its seat in the cylinder head.

The valve guide is made of grey cast iron and the valve seat is made directly in the cylinder head when the material is cast iron.

The camshaft actuates the valves. Its revolutions are half in the 4-stroke and full in the 2-stroke, with the rotations of the camshaft.

The underlying camshaft receives its drive from the crankshaft with toothed gears.

The head camshaft is mounted on chains, toothed belts or bevel gears with intermediate shafts and operates together with the camshaft. The camshaft is mounted on the head to prevent flexural oscillations of the valve system. This is necessary for high-speed engines to prevent vibrations in the valve train and springs. A broken valve spring causes great damage because the valve falls into the cylinder and destroys the piston. In chain and toothed belt drives, tensioners and drive guides are used to prevent oscillations and achieve a smooth, trouble-free transmission.

The material of the camshaft is forged or hardened steel or cast iron. The surfaces of the camshaft housing and camshaft bearing surfaces shall be hardened and ground.

The simple transmission mode (myceloid), (Fig. (5.9.1), Fig. (5.9.2) b). is used for the transmission of small forces. For larger forces, transmission is more easily achieved by means of cylinders, (Fig. (5.9.2) a), (Fig. (5.9.2) a), the construction of which is however more complex.

With the hydraulic mode of transmission, self-regulation of the valves is achieved.

The transmission mechanism is mainly made of cast iron. The surfaces of the cylinders are hardened and ground.

For good filling of the cylinder, the inlet cross-section must be large. The limits of the valve disc diameter are determined by the construction of the cylinder head and combustion chamber. The outlet valve is most often manufactured with a smaller diameter than the training inlet valve.

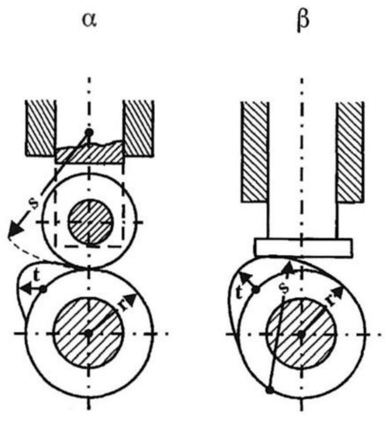


Figure (5.9.2) Modes of transmission mechanism.

A large intake valve allows for better cylinder filling. Also, during assembly it is impossible to confuse the valves, which is important because the exhaust valve is made of better material. In parallel valves the following disc diameters are chosen, valves of the same size d=0,4D, different valves, din= 0,45D, dex=0,35D, (D= cylinder diameter). The size of the valve stem diameter is in high speed engines dstem. = (0,25 to 0,35) d. and in low-speed engines dstem. = (0,15 to 0,25) d. The angle of the base of the valve is preferably 45° and the width of application between 1,5 mm and 2,5 mm. For the calculation of the maximum ramming of the valve, it is assumed that the flow at the valve cross section and shortly after the valve cross section is of approximately the same magnitude. Based on experience (5.9.3), the maximum valve embolism is between d/6 and d/4.

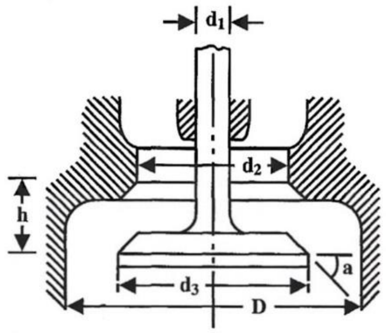


Figure (5.9.3) Valve construction details.

When the valve calculations are completed, the valve diameter and maximum embolism are checked based on the average flow velocity at the base of the valve. The average flow velocity, which is only a calculated quantity, is given by $u_{aver} \cos a \, \pi d_3 h_{max} = u_{piston aver} \, \pi \, D^2/4$, where $u_{aver} \cos a$ is the velocity component perpendicular to the cross section $\pi \, d_3 h_{max}$, $u_{piston aver}$ is the average piston velocity and D is the cylinder diameter. The average flow velocity is derived from the previous equation and compared with known values of similar engines. When its difference from the other quantities is large, the valve disc diameter and piston diameter must be redefined. Empirical values for the average flow velocity at the inlet valve are 60 to 90 m/s and at the outlet valve 80 to 120 m/s.

The shape of the cam is important for the valve travel and must meet the following requirements: a) the valve must open quickly to the maximum and close again just as quickly so that the flow through the cross-section lasts as long as possible, b) the travel path must be chosen so as to avoid impermissibly large oscillations (spring oscillations). Also, care must be taken to ensure that the natural frequency of the drive mechanism is as high as possible, using slightly rigid components. The cause of the oscillations is the camshaft and the first part of the acceleration of the camshaft. When the valve is closed, the transmission of force between the oscillating parts and the oscillation stops due to the valve gap.

The timings (timing angles) show how many degrees of crank angle the intake and exhaust valves are open. Because the valves open and close relatively slowly, the timing of the corresponding angle is chosen to be greater than the suction and exhaust timing corresponding to the piston. At the end of exhaust and at the beginning of intake they overlap, i.e. both valves are open at the same time. By covering the valves, the simultaneous removal (with the entry of the fresh mixture) of the exhaust gases from the combustion chamber is achieved. The exhaust gas mass flows due to its inertia even when the piston is already at the Top Dead Centre and therefore a vacuum is created in the combustion chamber by the suction of fresh mixture and the fresh mixture is sucked in through the already open intake valve. A large valve overlap provides better flushing of the combustion chamber and residual exhaust gases, but in Otto engines (except for direct injection engines), it leads to a loss of fuel consumption. An optimal solution is sought whereby the residual exhaust gases are removed as far as possible before fuel consumption is lost, especially under the current restrictions on environmental pollution from unburnt hydrocarbons. The intake valve is closed immediately after bottom dead centre so that, by taking advantage of the inertia of the fresh mixture mass, cylinder refilling is achieved. Refilling is faster the faster the fresh mixture flows in, i.e. the higher the average piston speed and the higher the engine speed. By delaying the closing of the intake valve, due to increased filling, we achieve high engine power at high rpm. Conversely, at low rpm the power is lower because the piston pushes back out part of the filling mixture. Figure (5.9.4) gives the torque and power as a function of rpm for faster and slower intake valve closures.

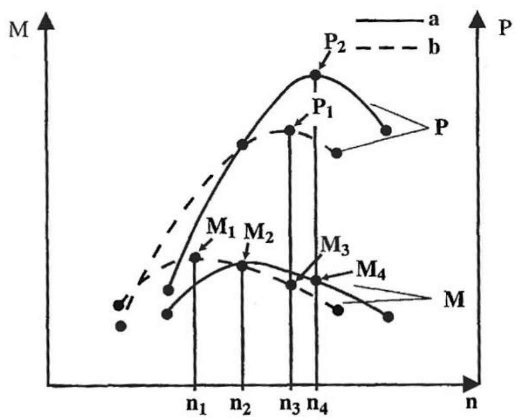


Figure (5.9.4) Torque and power versus speed for faster and slower closing of the intake valve, a, b slow and fast closing of the intake valve, P1, P2 power, M1, M2, M3, M4 torque and n1, n2, n3, n4 speed.

The elasticity of a high-power motor is low (the intake closes slowly). In this engine, more rpm is required to always be in the proper rpm range (the position of the drop in the characteristic torque curve between rpm that gives maximum torque and maximum power.

Timings are taken empirically, for intake valve opening 10 to 50° crankshaft crankshaft (CS) before (TDC), closing 40 to 80° CS after BDC, for exhaust valve opening 40 to 80° CS before BDC, closing 10 to 50° CS after TDC.

The harmonic cam consists of cylindrical sleeve surfaces, i.e. it is a cam of circular arcs. This circular arc cam cooperates with a drive mechanism and the generated motion can be described by simple harmonic relations. Then the cam of this type is also called harmonic.

The load switching in the two-stroke engine takes place during the time the piston is moving near the BDC. During this short period of time (100-150°C CS), the cylinder must be emptied and refilled with fresh charge (mixture). The piston here only plays the role of a guide for the intake and exhaust ports. The fresh mixture is sucked up by the compressor and fed into the cylinder at an overpressure of 0,1-0,4 bar. The overpressure also sweeps the exhaust gases towards the outlet. At the point where the exhaust gas and fuel mixture streams meet, mixing cannot be completely avoided. For load rotation, the air demand quantities (amount of fresh mixture per cylinder cycle in the cylinder, to the theoretical quantity under normal conditions), the cylinder absorption rate (actual fresh mixture in the cylinder, to the actual fresh mixture plus the remaining exhaust gases), and the retention rate (actual fresh mixture in the cylinder, to the amount of fresh mixture per cylinder cycle). The sweep (absorption) modes are defined according to the direction of the absorption flow in sweep in the same direction, reverse sweep and transverse sweep, (Figure (5.9.5). In the same direction sweep, the fresh mixture enters the cylinder through the intake port and exhausts the exhaust gases upwards through one or more exhaust stages. The intake ports are arranged tangentially to give the sweep air flow a rotary motion. During reverse sweep, the exhaust gases escape in the opposite direction to that in which they were from the incoming sweep air into the cylinder. The transverse sweep is the easiest to manufacture, but also has the smallest sweep effect. The sweep air stream is directed through the laterally mounted inlet ports up to the cylinder head. There, it changes direction and pushes the exhaust gases towards the outlet.

The sweep compressor introduces the fresh mixture with a slight overpressure into the cylinder. It performs the work performed by the piston in the four-stroke engine, namely, intake and exhaust. A rotary piston compressor, a reciprocating piston compressor and also a crankcase compressor are used to introduce the sweep air. Crankcase superchargers provide a minimum of sweep air and are therefore only used in Otio engines. In Oio engines with carburetors, the air demand rate must be such that the retention rate is approximately equal to unity. For a lower degree of retention, there would be losses not only of air but also of fuel.

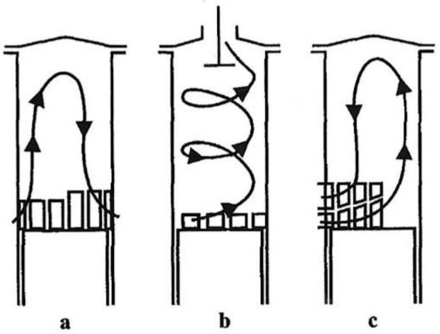


Figure (5.9.5) Scanning modes, a transverse, b same direction, c reverse scan.

CHAPTER 6

6 Supercharging

1.1 Supercharging

The fresh mixture in a turbocharged engine is fed into the cylinder, usually under supercharging, via a compressor. This increases the amount of fresh mixture (more fuel is burned) and the actual net power increases in proportion to this increase. Today, all large marine diesel engines are turbocharged. The scope of turbocharging has even extended to low power engines, e.g. truck engines. In Otto engines, the application of turbocharging has so far been slow. There are two main reasons for the application of turbocharging: a) the increase in useful power, with a simultaneous reduction in specific fuel consumption, when comparing engines of similar construction sizes at the same operating speeds: b) the equation of the continuously decreasing engine power with the increase in the altitude of the position or, respectively, the flight (increase in turbocharging).

A picture of the variables that play a role in the increase in useful power is given by the relationship Pe = peVhzni. When transferring from the four-stroke to the two-stroke engine, based on the above formula, we had to lead to a doubling of power. However, this is not possible because the average effective piston pressure pe is lower in the two-stroke engine than in the four-stroke engine. Usually, it is not possible to increase power by changing the duty cycle, because other variables are important for its selection. In an increase in displacement Vh, or number of revolutions n, the average piston speed uμ must not be exceeded and also the optimum piston ratio and must not be changed. Based on these constraints, it follows that it is possible to increase either the displacement, or the number of revolutions, or to change the number of cylinders and the number of revolutions, or the number of cylinders and the displacement. By combining the quantities Vh, n, k and uμ. Where:

$$V_h = \frac{\pi D^2}{4} H$$

$$\kappa = \frac{H}{D}$$

$$u_\mu = 2\mathrm{Hn}$$

We have:

$$V_h = \frac{\pi}{32}\frac{1}{\kappa^2} = \frac{u_\mu^3}{\pi^3} \qquad (6.1.1)$$

In relation:

$$P_\varepsilon = p_\varepsilon V_h z n i$$

We want to double the power. The increase in power must be achieved by increasing pε, Vh, z and n. The quantities i, k and uμ remain constant. To double the power, we double in order the average effective piston pressure, the displacement, the number of cylinders and the number of revolutions. Increasing the net power payload by increasing the average effective pressure or the number of cylinders is possible without changing any other variable. Increasing power by increasing the number of revolutions is neither possible nor practical, because then the number of cylinders (and displacement) would have to be increased by the third power of the power ratio. Increasing the displacement or number of cylinders is only possible when there is sufficient installation space, taking into account the increase in engine weight. If this solution is not possible, the only remaining option is to increase the power by increasing the average effective piston pressure.

The equation gives information on the possibility of increasing the average effective piston pressure:

$$p_e = n_{th} n_g n_m \lambda_1 H_\alpha \frac{\rho_\alpha}{\rho_o}$$

The thermal degree nth increases with the compression ratio e. In the Otto engine, the operating limits are limited in relation to the compression ratio due to the need for smooth engine operation without knocks. The Diesel engine already operates at a high compression ratio and a further increase would reduce the mechanical efficiency instead of increasing the thermal efficiency. With early ignition timing and a reduction in flow losses during load switching (e.g. by smoothing the gas channels), the ng combustion quality grade can be increased slightly. And in the diesel engine with better combustion quality, it results in an improvement of the ng quality grade. The mechanical quality grade nm is improved when bearings are used in the crankshaft instead of rolling bearings. The calorific value of the mixture Ha is determined by the calorific value of the fuel and the required air ratio. What remains, therefore, as the main active means of increasing the average effective piston pressure is to increase the flow rate λ1. The degree of delivery λ1 is increased by means of supercharging.

To increase the mixture delivery ratio, i.e. supercharging the engine, there are the modes of external supercharging, mechanical supercharging, exhaust gas supercharging and pressure oscillation supercharging. In the first three processes, a compressor drives the pre-compressed fresh mixture into the cylinder, while in the latter case, gas pipe oscillations are used to fill the cylinder with overpressure.

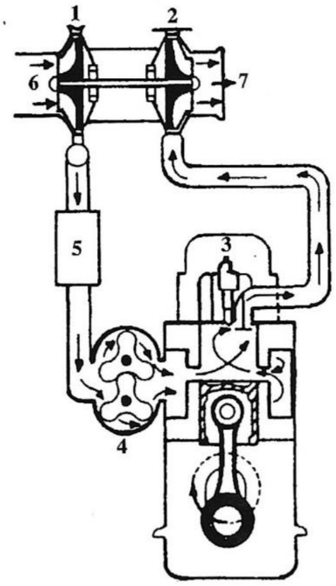


Figure (6.1) 2-stroke Diesel with turbocharger and mechanical rotary piston compressor, 1 compressor, 2 turbine, 3 injector, 4 rotary piston compressor, 5 intercooler, 6 air, 7 exhaust gas.

External overfilling and mechanical overfilling differ from each other in the function of overfilling. In external supercharging, a special motor, e.g. an electric motor. The amount of supercharging supplied can be adjusted to the required degree of supercharging, regardless of the engine speed. Such a compression unit is accurate. And that is why it is found today at most as an additional compressor in two-stroke turbocharged engines. Mechanical supercharging, in which the compressor is driven directly by a trapezoidal belt, chain or sprockets, is simple to operate and relatively inexpensive. Mechanical supercharging is mainly used in small engines, because in these engines turbocharging would significantly increase the price of the turbocharger. Also, gas turbochargers operate at low gas flows with low efficiency.

The most used compressors are those of rotary piston compressors (Roots-compressors). The flow rate of these compressors varies with speed and is almost independent of backpressure, and because the suction capacity of the engine increases with speed, the rotary piston compressor is the most suitable for supercharging. Figure (6.1) shows a two-stroke diesel engine with a turbocharger and a mechanical rotary piston compressor.

Fig. (6.2) shows the p, V diagrams of a simple engine and a turbocharged engine. The piston of the turbocharged engine produces positive work also during cylinder filling. This recovers the energy that the compressor delivers to the fresh mixture.

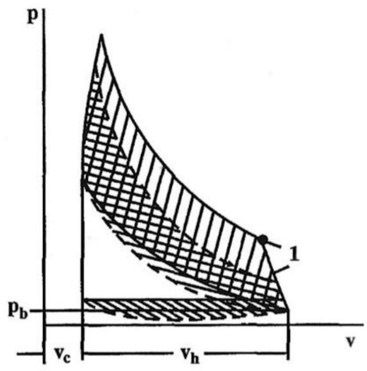


Figure (6.2) Plots p, V in a naturally aspirated supercharged and turbocharged engine, 1 exhaust open.

With an open exhaust, the cylinder pressure in a turbocharged engine is higher than in a normal engine and more energy is lost with the exhaust gases. The specific fuel consumption, especially in the engine operating range, is higher in a mechanically supercharged engine than in a non-turbocharged engine, when comparing the same constructional dimensions. This comparison is not entirely correct because the two engines have different net power outputs. Increasing the sizes (displacement, number of cylinders) of the non-turbocharged engine to give the same net power as the turbocharged engine increases the friction power disproportionately to the power losses due to the turbocharger. Thus, on a more accurate basis of comparison, the turbocharged engine may have better specific fuel consumption.

In turbocharging, the engine exhaust gases do not leak directly into the environment, but first deliver their energy (which is particularly high in turbocharged engines) to an exhaust gas turbine. The exhaust gas turbine drives the compressor mounted on the same shaft. The compressor sucks in the fresh mixture and forces it under overpressure into the cylinder (Fig. (6.1). The turbine and compressor assembly is called a gas turbine compressor. In multi-cylinder engines, more ducts lead to the gas turbine, (Fig. (6.3)).

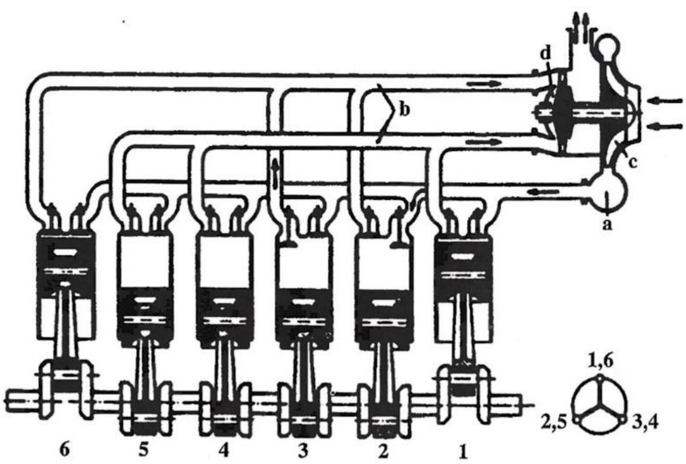


Figure (6.3) Exhaust gas turbine-compressor and exhaust system. a fresh air, b exhaust gas, c the compressor, d the turbine, ignition sequence 1-2-3-4-6-5-3, exhaust gas duct communication 1-4-5 and 2-3-6.

In exhaust gas turbocharging, the charging unit is connected to the engine only via the gas stream. Therefore, the question arises as to how quickly the compressor reacts to the change in engine power. With an increase in power, by increasing the injected quantity (diesel engine), there is initially less air for the corresponding amount of injected fuel. This can lead for a short time (about 1 sec) to smoke production (poor mixture conditions). The exhaust gas temperature also rises for a short time above normal until the gas turbine, which receives more energy through the hot exhaust gases, accelerates the compressor unit to a new operating state. The turbocharged engine reaches idling almost as quickly as the non-turbocharged engine.

The turbocharger compresses the fresh mixture. This increases not only the pressure but also the temperature. Therefore, the density increases depending on the pressure and the filling of the cylinder. The cylinder fills with less mass than would correspond to the net increase in pressure. This reduction in mass, due to the increase in temperature, is particularly noticeable at high compression ratios. Therefore, in high turbines, the air after the compressor is cooled in an intermediate cooler before entering the cylinder. The cost of the compression unit naturally increases with the addition of the air cooler, but benefits are reaped by increasing the feed mass of the fresh mixture and the relatively cold fresh mixture reduces the cylinder temperature. If a heat balance is done on a supercharged engine, it turns out that the heat dissipated by the cooling water is less than in an engine without supercharging. On the contrary, the heat extracted per kWh increases in the exhaust gases because the cylinder is very well swept.

The four-stroke diesel engine can be operated with both shock turbocharging and delay turbocharging. Additional compression is not necessary. There is no difficulty in removing the exhaust gases from the cylinder because the piston pushes the exhaust gases outward, despite the back pressure due to the gas turbine. Combustion is not as fast because the ignition delay time is reduced due to the increased oxygen concentration. Low-high pressure supercharging is distinguished. The limit between them is about 50% increase in power.

In the case of low-pressure supercharging, the normal non-supercharged engine can also be used. The valve overlap is 100 to 140° CS, so that the compression chamber is very well swept, and temperatures remain low. Through good internal cooling with air sweeping, the exhaust gas temperature in the supercharged engine is not much higher than the temperature of the non-supercharged engine.

At a power increase of more than 50%, we are talking about high pressure overfill. The engine must be built for conditions of maximum mechanical and thermal stress. Ignition pressures occur up to 120 bar. At a doubling of power, the compressor pressure increases to about 2.5 bar. The limit for supercharging without intercooling is approximately at a compressor pressure of 1,8 bar and a temperature after compression of 110 °C. Therefore, an air cooler must be used in high pressure supercharging. The air temperature inside the air cooler should reach 50 to 60 °C. The engine cooling water passes through the air cooler. The exhaust gas temperature is limited by the permissible turbine temperatures of 600 °C for continuous operation and 650 °C for intermittent operation.

The method of supercharging the exhaust gases in two-stroke diesel engines has been delayed for a long time. The two-stroke engine had no interest in increasing its power because the four-stroke engine was superior anyway. Problems had to be overcome. such as high thermal stress on the piston and cylinder. In the two-stroke engine, maximum power is determined not only by the soot emission limits but also by the maximum permissible piston temperature. New supercharging methods must therefore be developed so that the thermal stress remains practically the same as in the non-supercharged engine. To keep the thermal stress low, the air is cooled before it enters the cylinder (after the compressor). Supercharging the two-stroke engine is a difficult task because the exhaust gases are not pushed by the piston towards the outlet, but by the fresh mixture entering the cylinder. Therefore, the cylinder sweep pressure must always be higher than the exhaust gas pressure before the gas turbine. This requirement is very difficult to achieve, especially in the two-stroke engine. In two-stroke turbocharged engines, the process is assisted by additional compressors, which are usually installed in parallel and in series with the exhaust gas turbocharger. The power increase in the two-stroke engine is about 30%.

So far there are few Otto engines with exhaust gas turbocharging. A more powerful engine is preferable to a small, turbocharged engine. The turbine and compressor shafts must have a small diameter because the gas leakage volumes in small displacement Otto engines are small. Such small-sized wheels naturally operate with low efficiency. In Otto engines the ignition pressure increases approximately in proportion to the compressor pressure. The limits of the supercharger are selected so that no knocking (pins) occurs. To reduce the risk of knocking, the compression ratio is usually reduced.

When the throttle valve is fully open, supercharging begins at half the rated engine speed, because only then does the turbine have the required amount of exhaust gas. The compressor pressure increases rapidly with engine speed and already reaches 3/4 of the nominal speed, the maximum permissible value. It is observed that, through the supercharging of the exhaust gases, a not so good torque curve is obtained, (Fig. (6.4). To improve the torque curve (at low speeds the torque should be high and at high speeds it should drop), the turbocharger unit must be adjusted. Usually, when the maximum allowable compression pressure is reached, a quantity of exhaust gas bypasses the turbine and goes directly to the exhaust.

The carburetor can be installed before or after the compressor as a pressure carburetor. However, engines with carburetors have the disadvantage that they cannot sweep the compression chamber efficiently (you would have very high fuel losses). In gasoline injection engines, as in diesel engines, the compression chamber can be well swept so that the engine thermal stress and exhaust gas temperature can be more easily kept within the permissible limits. To avoid fuel losses during sweeping, the fuel must be injected when the exhaust valve is closed, i.e. for each cylinder the injection valve must be adjustable.

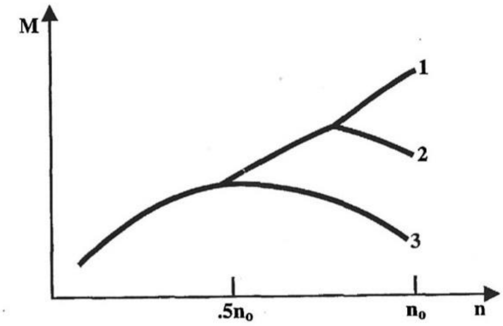


Figure (6.4) Typical Otto engine torque curve, 1 turbocharged engine without regulation, 2 turbocharged engines with regulation, 3 turbocharged engines without regulation.

The change in load (fresh intake-exhaust mixture) in an engine is reflected in the pressure oscillations that occur. By exploiting these oscillations, we achieve a supercharging effect. To achieve this, the frequency and phase position of the oscillation must be matched to the frequency of the engine operating cycle. A good cylinder filling in the four-stroke engine is achieved under conditions of low cylinder pressure, at the end of the exhaust valve opening period (so that very little exhaust gas remains in the cylinder) and at the closing of the intake valve the cylinder pressure must be high. By properly determining the length of the pipes and their cross-sections, it is possible to meet the above requirements. And the two-stroke engine can be turbocharged without a sweep turbocharger by sizing its pipelines. Coordination, through sizing, can only be done for a small speed range because the gas mass coordination depends on the length of the ducts.

ΣΥΜΠΕΡΑΣΜΑΤΑ

Στο πλαίσιο της παρούσας Διπλωματικής Εργασίας, μεταφράσαμε το σύγγραμμα του κυρίου Μαυρίδη Κωνσταντίνου ‘’Μηχανές Εσωτερικής Καύσης’’. Αυτό το σύγγραμμα μας βοήθησε να κατανοήσουμε καλύτερα, τον τρόπο με τον οποίο λειτουργούν οι μηχανές εσωτερικής καύσης, τις οποίες χρησιμοποιούμε σε μεγάλο βαθμό στην καθημερινή μας ζωή. Μέσω της μετάφρασης, καταφέραμε να διευρύνουμε τις γνώσεις μας επάνω στις Μ.Ε.Κ., πράγμα που θα αποτελέσει μεγάλη βοήθεια στην επαγγελματική μας πορεία ως μηχανολόγοι μηχανικοί. Επιπρόσθετα, αναλύοντας βασικά κομμάτια της λειτουργίας τους, διαπιστώσαμε πόσο σημαντικές είναι κάποιες πληροφορίες σχετικά με την σωστή χρήση και συντήρηση των Μ.Ε.Κ. . Κλείνοντας ελπίζουμε πως αυτή η εργασία και κατ’ επέκταση η μετάφραση του συγγράμματος, θα βοηθήσει φοιτητές προερχόμενους από το πρόγραμμα ERASMUS, οι οποίοι δεν μιλούν Ελληνικά ή και οποιονδήποτε άνθρωπο ή φοιτητή που τον ενδιαφέρει το θέμα αυτό, να διαβάσουν με ευκολία και να κατανοήσουν το βιβλίο αυτό, μεταφρασμένο στην διεθνή γλώσσα και να γίνει χρήση από φοιτητές του Πανεπιστημίου οι οποίοι σπουδάζουν ή έχουν σκοπό στο μέλλον, στο εξωτερικό.

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