

きた。このボイラは過熱部改造後は  $138^{\circ}\text{C}$  以下の過ガス温度で運転されているが空気予熱器にひどい損耗は起していない。なお燃焼ガス中の固形可燃分の割合を測定する装置についても説明してある。〔山崎 康雄〕

621.43.008

※(566) 燃料噴射 (E.M. Goodger, *Auto. Engr.*, 1956—3, Vol. 46, No. 3, p. 110—113, 図 6) 大花点大機関の気化器の代りとして用いられる燃料噴射装置数種類について説明している。まず従来の気化器の原理から話を進め浮子式気化器の特徴、その欠点を補うため二つのダイヤフラムを用いた航空機関用の Bendix-Stromberg 圧力気化器の原理、機関の消費する空気量が吸入管の空気密度と機関の回転数に比例することを基礎とした速度—密度調整噴射装置、回転斜板とノズル機構によって燃料の量を加減する S.U. 燃料噴射ポンプ、ディーゼル機関に用いる Bosch 燃料噴射ポンプに吸入圧力とダイヤフラムとを釣合せる補助機構を用いた装置、二つの らせんみぞ で噴射時間を定める Hobson 燃料ポンプなどを詳述している。

結論として燃料噴射は均一な混合気を各シリンダに分配するため出力の増加、起動のしん速、バックファイアの除去などの利益があげられるが、反面欠点として過給機関の燃料噴射における燃料冷却の不足と燃料噴射装置の高価と複雑な点を指摘している。レース用自動車機関には速度—密度調整噴射装置が航空機関には速度—密度—背圧調整が適しており、大量生産に対する簡素化と今一步の性能向上が望まれている。

(桑田 寅男)

621.436.1—222.2 : 539.373.6 : 621.43.016.4

※(567) ディーゼル機関のシリンダとピストンリングの摩耗に及ぼす表面温度の影響 (H.V. Nutt 外 2 名, *S.A.E. Trans.*, 1955, Vol. 63, p. 694—703, 図 20, 表 1) ディーゼル機関のシリンダとピストンリングの摩耗の中、磨食は潤滑油にアルカリ性添加剤を混入することによって防ぐことができるが、接触摩耗はシリンダ内壁の高温のため潤滑油膜が蒸発し破壊されることによって生ずるので、これを防ぐには燃焼ガスの温度、流れ、ライナの熱伝導度、冷却水の温度と湿度、排除燃気の量と温度等の因子から壁温を計算してそれができるだけ低くするような設計をすればよく、また一方潤滑油の方でも壁温を減らすよう粘性と同時に揮発性も考えねばならない。他の条件を同じにし、負荷を  $1/2$  時間に  $1\text{ kg/cm}^2$  (平均有効圧力) ずつ増して放射能リングによる実験を行ったところ、ある負荷までは低い割合で摩耗が増すが、あるところで急に増加する。この点を油膜破壊の負荷とし、これと各油の特性  $V$  ( $300^{\circ}\text{F}$  における SUS) と、揮発度  $P$  (加熱後の残留分%) の関係を求めると、個々の因子だけでは必ずしも破壊負荷の傾向と一致しないが、これの一次結合

$$F = -2 + 0.3P + 0.9V$$

なる係数  $F$  を作ると、その破壊負荷の良否順序とよく一致することがわかった。〔斎藤 武〕

621.436.1.019

※(568) 小形高速ディーゼル機関の所要燃料 (G.M. Barrett & H.G. Preston, *Insta. Mech. Engr.*,

*Proc. (Automobile Division)*, 1954—55, No. 8, p. 218—224, 図 11, 表 1) シリンダ体積  $1.0\text{ l}$  以下、最高回転数  $2400\text{ rpm}$  くらいの小形高速ディーゼル機関では、燃焼をオートサイクルのそれに近づける必要があり、噴射期間は  $20^{\circ}$  以内に短くするのがよい。ただし磨食と振動は多くなるので、これは両面式燃焼室を用いるとかして別に考究せねばならぬ。ポンプは従来の直列式より分配式がよく、速度に応じた時期調節装置を必要とする。  $2500\text{ rpm}$  直列噴射機関の例によると、セタン価は燃料消費量には明確な差異を示さず、可燃燃発生負荷がセタン価の大きいほど高い程度、ただし着火遅れは少いが、これは自動時期調節装置をつければ解決される。冷燃効は着火性と逆の傾向で両者を考慮するとセタン価  $55$  くらいの燃料が適当。滞留点は決選、デボットが少ないよう低い方がよいが、貯点はあまり低いと噴霧に貫通性なく不適当。また高気圧が高いと高気閉をくを起し易い。比重  $0.80\sim 0.85$  の範囲では燃費の差違  $3\sim 5\%$  でごくわずかである。〔斎藤 武〕

621.436.1 : 629.113.52

※(569) ディーゼル機関を軽車両に用いるには (N. M.F. Villiamy, *Insta. Mech. Engr.*, *Proc. (Automobile Division)*, 1954—55, No. 8, p. 225—245, 図 15, 表 3) 乗用車等の路上軽車両用に設計試作したシリンダ体積  $3.4\text{ l}$  以下の小形ディーゼル機関に関する実験的研究。試作機関は直径  $73.0$ 、行程  $93.9\text{ mm}$ 、行程体積約  $3.8\text{ l}$ 、シリンダ数は  $1$  および  $4$  の 2 種、四サイクル、頭上弁、頭上カム・チェーン駆動式、圧縮比  $19$ 、燃焼室は渦流室のややのびた形、最大平均有効圧力  $7.9\text{ kg/cm}^2$  ( $1800\text{ rpm}$ )、最高回転数  $4000\text{ rpm}$ 、水冷、かわきライナ使用、クランク軸はガソリン機関のを転用、燃料ポンプは  $5\text{ mm}$  ブラッジャ、ピントルノズル、噴射圧力  $125\sim 100\text{ at}$ 、小噴霧開角。高噴射率のものが良い、静的噴射始め角度を自動進角装置によって  $-17^{\circ}$  ( $400\text{ rpm}$ ) から  $-35^{\circ}$  ( $4000\text{ rpm}$ ) に変え、実際の噴射始め角度がほとんど一定になるようにする。ポンプの摩耗、サーシングの間隔は材料と設計によって改良され、機関の最大トルクがポンプ回転で  $1000\text{ rpm}$  になるよう調節するのがよい。噴射弁リフトを高く、スプリング初荷重を強くして弁のおどりを防ぎ ( $4400\text{ rpm}$  まで見られず) はだ焼割タペットを使用して磨食性をよくした。この機関を  $1.5\text{ l}$  のガソリン機関自動車にとりつけたところ、高速時の加速性よくトップスピードも  $72\text{ mph/4000 rpm}$  に増した。ミッション、クラッチはもとのまま、燃費  $0.216\text{ lb/bph}$  ( $2000\text{ rpm}$ )、 $0.267\text{ lb/bph}$  ( $4000\text{ rpm}$ )。以上を同じ出力のガソリン機関と比べると、車の速度、機関出力、大きさ、重量等では十分の性能をえられるが、静粛さ、アイドリング時の振動、排気等の点ではさらに研究を要する。〔斎藤 武〕

621.436.1.019.2

†(570) 四および二サイクル・ディーゼル機関のターボ過給に関する熱流体的考察 (R. Bärmann, *A.S. M.E. Trans.*, 1956—1, Vol. 78, No. 1, p. 171—183, 図 8) 実験と理論的計算を基にして、種々のターボ過給圧力比に対する平均有効圧力、燃焼比、熱

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