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HYDROGEN ENERGY CONVERSION

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July 1976

Semi-Annual Technical Report
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Air-breathing reciprocating engines and gas turbines fueled with hydrogen are treated with special concern for the problems each might encounter if used on a large scale. The potential improvements in performance of each power plant when operated with hydrogen are determined using prior analytical and experimental data and/or new estimates. Particular attention is given to factors which might improve conversion efficiency and discount, at least partially, the relatively high cost of hydrogen energy.

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A previously reported summary of prior data from naturally-aspirated, spark-ignition engines is evaluated farther in light of new theoretical ("fuel/air cycle") calculations and found highly consistent with them. Though based on fewer experimental data of less scope, a similar comparison is made between experimental and analytical results for hydrogen engines with direct-fuel-injection. Problems cited include fundamental limitations on power output with natural aspiration as well as insufficient experimental data especially with fuel-injected engines. Mapping of naturally-aspirated engine performance and further development of combustion control by fuel injection are recommended.

Previous operating experience and analysis of gas-turbine operation on hydrogen are considered. Hydrogen-fueled gas turbines are found, unlike reciprocating engines, to offer relatively modest thermodynamic performance gain compared with hydrocarbon fueling, though LH_2 fueling has substantial potential (undemonstrated) for power-plant efficiency or reliability improvement through hot-section cooling and/or heat regeneration. LH_2 fuel-system problems with transient response as well as inavailability of suitable hardware are most evident, though probably resolvable by substantial state-of-the-art development.

For both power plants, long-term operating data and operational-safety hazard experience (especially in enclosures) are lacking, and this subverts evaluation as to ultimate practicality.

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TABLE OF CONTENTS

EXECUTIVE SUMMARY

TECHNICAL PROBLEM

The work reported here is part of a larger effort concerned with the possible large-scale use of hydrogen as a fuel. This effort is aimed at (1) identifying and defining the technical problems associated with such use, and (2) indicating solutions or approaches to solving such problems. At its initiation in January 1974, the program effort was to cover hydrogen production, storage and handling and energy conversion. During the program, hydrogen production was deemphasized by mutual agreement. Likewise, early in the program, hydrogen furnaces, catalytic burners, and fuel cells were precluded from consideration as energy-conversion devices.

The present report is Volume II of the Third Semi-Annual Technical Report for the subject program. Volume I was prepared previously.* The intent of both of these reports is to provide documentation for much of the program's Final Technical Report. Therefore, the present volume emphasizes problem identification although routes to the solution of various key RD&D problems are called out. The final report will emphasize synthesis and a broader viewpoint in treating hydrogen-use problems and solutions.

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*Cole, R.B., Magee, R.S., and Hollenberg, J.W.; "Hydrogen Storage and Transfer", Report ME-RT-75008 (to ARPA under Contract N00014-75-C-0220), Stevens Inst. of Tech., August 1975, 79 pp.

GENERAL METHODOLOGY

The present report on hydrogen energy conversion represents two major efforts, one concerned with hydrogen-fueled reciprocating engines and the other with gas turbines.

These two specific classes of power plants were emphasized at the expense of concern for more visionary power-plant concepts for two reasons (which will be treated further in the final report for this program). First, these are the most common present-day power plant types and, considering capital investments and operating experience with them, they are most likely to maintain in the near future (e.g., 25 years). Therefore, any large-scale, near-future use of hydrogen must necessarily impact substantially on these widespread energy-conversion devices. Second, there is a need for practical experience with hydrogen in commercial, industrial and military situations more presentative of large-scale use than are the rigorously controlled and monitored situations of hydrogen use by NASA and industry in the past and present. A major demonstration of a hydrogen-fuel system is desirable for the near-future, and this should necessarily involve reciprocating and/or gas-turbine engine.

Reciprocating engines were investigated most strenuously in the early stages of the subject program, and these results were previously reported.* Early emphasis on reciprocating engines was justified on the basis of immediate availability of a large body of experimental data which had never been overviewed and evaluated. The present report adds to the previously reported work in two regards: (1) some additional, experimental data have been incorporated (particularly for cylinder-fuel-injected engines) and (2) the results of theoretical performance calculations are incorporated.

Hydrogen-fueled gas turbines were investigated in detail later in the subject program. A smaller body of experimental data exist in this area, and the literature in this area is less immediately

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* McAlevy, R.F.III, Cole, R.B., Hollenberg, J.W., Kurylko, L., Magee, R.S., and Weil, K.H.; "Hydrogen as a Fuel", Report ME-RT-74011 (to ARPA under Contract N00014-67-A-0202-0046), Stevens Inst. of Tech., Aug. 1974, 235 pp.

accessible, being comprised largely of NACA reports of the late 1950's.

Since the gas turbine is conceptually a multi-fuel power plant, performance estimates for hydrogen were made by comparison with conventional hydrocarbon-fuel performance rather than in absolute terms. The potentials for significant gains in performance were identified in order to focus concern on hydrogen-fueling problems. Those areas were identified (e.g., improved hot-section cooling) which offered the most advantages to trade-off against the known major disadvantages of hydrogen fueling (production costs, storage volume, and uncertainty of safety).

In addition, problems related to gas-turbine fuel-system operation were focussed on. Numerous reports (in the literature and otherwise) indicated substantial practical problems with fuel pressurization and control. In addition, the likelihood of liquid hydrogen (LH_2) use in marine and air-borne applications led to concern for problems in LH_2 pumping and heat transfer (vaporization).

Beside use of the published literature, extensive consultation with experienced hydrogen users and researchers was employed in this program. Numerous technical discussions were held with current and past reciprocating-engine investigators and with gas-turbine manufacturers (e.g., Pratt & Whitney) and hydrogen gas-turbine users (e.g., NASA, U.S.A.F.)

TECHNICAL RESULTS

Reciprocating Engines

H_2/O_2 reciprocating engines, while unique in several respects, are undeveloped and unlikely to see large-scale application.

Compression-ignition hydrogen engines hold little promise, except as dual-fuel engines employing pilot charges of hydrocarbon fuel.

H_2/air-modifications of conventional spark-ignition engines operate satisfactorily at low output power levels by means of lean burning and power-control through mixture-ratio variations (rather than throttling). The relatively high efficiencies and low NO_x emissions which are typically reported for such lean burning can be correlated on theoretical grounds, and even incomplete performance data from diverse sources can be consistently rationalized theoretically. A comprehensive, experimental "mapping" of H_2/ air vis-a-vis gasoline/air operation is not, however, available from previous experimental work.

Lean burning of hydrogen is advantageous due to the resulting high efficiencies (e.g., 50% higher than with gasoline) and low NO_x emissions (e.g., 1/10 to 1/1000 of those with gasoline) at comparable low loads. Lean burning, while yielding efficiency and emissions benefits, is necessarily accompanied by a specific power-output* penalty (relative to gasoline) of 40 to 50% or more. This is a major problem** and is inherent if high efficiency and/or low NO_x emissions are to be achieved. The power-output penalty can be overcome by dual-fuel operation (hydrocarbon at high load, hydrogen at low load) or by hydrogen supercharging ("direct"- or "cylinder-fuel-injection"). Cylinder fuel injection can theoretically result in a maximum-power advantage for H_2/air relative to hydrocarbon/air. To date, this potential advantage has not, however, been demonstrated.

Several H_2/ air engines have been operated with cylinder fuel injection ("CFI"), but testing and development of this type of hydrogen engine are insufficient to allow definitive evaluation of it. Theoretically, a maximum-power gain of 10 to 12% is possible relative to a comparable hydrocarbon/air engine. Experimentally, the highest specific power output for a CFI engine is still about 10% below that of a comparable gasoline engine. CFI engines, however, have demonstrated power output levels nearly 30% higher than comparable non-fuel-injected hydrogen engines even while operating lean to avoid rough-running.

Richer, near-stoichiometric (peak-power) operation of CFI engines has in some cases been forestalled by mechanically troublesome "rough-running". In other cases, near-stoichiometric operation has been reported but with power outputs so far below theoretical as to suggest combustion inefficiency. It is not clear that CFI engines

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* e.g., power/displacement

**the only other fundamental problem identified is that of safety can maintain high efficiency (or low NO_x emissions) at high power output, for example, by the usual "fixes", i.e., exhaust-gas recirculation or water injection. The major fundamental problem with CFI H_2/air engines is a lack of experience which demonstrates this engine's potential advantage in power output while achieving reasonably high efficiency and (less likely) low NO_x emissions through fuel-injection and combustion control.

More practical, less fundamental problems (aside from hydrogen production and storage and logistics problems) include:

(i) lack of long-term operating experience (uncertain reliability and maintenance),

(ii) lack of experience in conversion of existing practical Diesel-engine hardware to effective operation with hydrogen,

(iii) need for state-of-the-art advances in H_2-gas compressors and H_2-gas injection equipment (for CFI).

Gas Turbines

Being inherently multi-fuel engines, gas turbines should encounter no fundamental problems with hydrogen fueling*. However, for the same reason, hydrogen presently offers little technical advantage to gas-turbine performance except decreased emissions of carbon compounds and decreased hot-section corrosion problems (if sulfur-free hydrogen is used).

Between industrial and vehicular gas-turbine applications, hydrogen is only especially favorable for aircraft, but this is due to its high heating value (per lb.) rather than its performance in gas turbines. Otherwise, there is no unique value in hydrogen fueling except decreased carbon-compound emissions; performance gains are modest.

Between large-size and small-size gas turbines, hydrogen is favored more in small units. In small gas turbines, hydrogen's

* excepting safety and materials compatibility, perhaps. No materials problems are apparent from prior hydrogen gas-turbine experience (over very short test periods).

potentials are relatively more advantageous: increase thermal efficiency, decreased emissions, and (with LH_2) decreased turbine cooling-air requirements. However, LH_2 use with its increased fuel-system complexity and size is not as favorably joined with a small gas turbine as with a large one.

Future development to provide high-temperature fuel-injection, increased temperature uniformity at combustor outlet, and/or improved hot-section cooling offer potential for modest improved engine efficiency (e.g., 5 to 15%) over hydrocarbon-fueled turbines. This derives from a potential for increased turbine inlet temperature. There are no major problems apparent if such development were undertaken, though if high-pressure hydrogen is not available from storage, fuel-gas compression is a substantial practical problem. Costly, sizeable, special equipment of questionable reliability for gas compression.

No single hydrogen-fueled gas-turbine is known to have been operated over about 20 hours, and no complete powerplant has apparently ever been designed, fabricated, and used (or tested) in a fully operational mode.

Cooling of gas-turbine hot sections using (indirectly) the heat-sink capacity of cryogenic liquid hydrogen ("LH_2") is attractive and offers the largest potential for improvement of gas-turbine engine performance. Efficiency increases of 10% or more are possible, again by way of increased allowable turbine inlet temperatures. However, the use of LH_2 is accompanied by a variety of practical problems, none fundamental (except safety), but all representing needs for substantial engineering development. LH_2 pumping and vaporization are significant problem areas especially because of the large variation in fuel-flow rates required in many gas-turbine applications and because of the desirability of fast start and transient response in many situations as well as the need for over speed protection. Appropriate pumps are non-existent and probably require major development programs, particularly if aircraft applications are involved; suitable flight-weight pumps have apparently never been designed or produced. Workable vaporizers have been produced and tested but not over extended life-times or in full-range operational systems subjected to typical fuel-flow turn-down ranges. Fuel-flow control problems (e.g., flow and pressure oscillations) have been common in previous, limited-scope investigations of LH_2-fueled gas turbines, and clear choices among alternative control schemes will depend on substantial control, design, and development efforts in the future.

LH_2-fueled gas-turbines have the potential of operating in combination with auxiliary-power heat engines using the LH_2 turbine fuel as a heat sink or as the working fluid; the former is preferable thermodynamically. While extensive parametric studies of this alternative were not performed, rough calculations indicate a potential for recovering close to 50% of the ideal energy input required to produce LH_2, adding about 10% to the gas-turbine power output and, hence, to its efficiency. Such a gain is modest compared with the actual energy cost for H_2 liquefaction (e.g., 15%) and overall power-system efficiency (including liquefaction) are still low (e.g., 15-20%). Substantial engineering design and development would be required.

Safety

Despite an extensive data bank concerning hydrogen's hazardous properties, significant operational-safety data is scant. A major reason for this is that large-scale use of hydrogen (especially LH_2) has to date been in carefully controlled situations and generally not in enclosures. Hydrogen use in enclosed quarters or engine compartments (e.g., marine, aircraft, land-vehicle) carries a substantial risk including, at the worst, detonation. Experienced users emphasize that good prior safety experiences with hydrogen are a result of extreme care in design, operating procedures, personnel training, etc., and that wide-scale use by inexperienced personnel cannot be expected to maintain a low accident frequency. Considering the large volumes required for typical hydrogen-fuel storage, armoring and other military precautionary measures are likely to be very compromising of functional design. Though not developed sufficiently at present, metal hydrides are relatively promising media for safe storage of engine fuel for land-based powerplants. Hydrides, however, carry a profound weight and/or cost penalty at their present stage of development.

IMPLICATIONS FOR FUTURE RESEARCH

Open questions regarding hydrogen as a reciprocating-engine or gas-turbine fuel have been identified in the subject study. However, for the most part, these are not fundamental, research-oriented questions; the unresolved questions in hydrogen-fuel use with air-breathing engines are largely at the development and demonstration end of the RD & D spectrum.

Several engineering research efforts should, however, be undertaken in the near future. Hydrogen production costs are anticipated to be high. Therefore, potentials of hydrogen for increased energy efficiency must be well established to factor into meaningful alternative-fuels studies. To establish better hydrogen's fundamental potential vis-a-vis other fuel alternatives, the following researches are recommended:

Reciprocating Engines

(1) A full experimental "mapping" of hydrogen's performance in conventional, naturally-aspirated reciprocating engines. Presently available data are not sufficient in scope to provide a clear-cut basis for comparing hydrocarbon-fueled and hydrogen-fueled engines. This is particularly true at low part-load conditions where hydrogen enjoys an efficiency and emissions advantage. Such data are imperative for realistic appraisal of the energy savings available in hydrogen in vehicular applications.

(2) The small base of experimental data on cylinder-fuel-injected hydrogen engines must be augmented. Experimental research investigations of fuel injection as a means of combustion control is imperative if the 20% penalty in maximum power of naturally aspirated hydrogen engines is to be avoided and at the same time NO_x emissions are to be minimized and efficiency maximized over a wide range of output power. High combustion efficiencies and practically low rates-of-pressure-rise and low NO_x emissions have yet to be observed in high-output-power hydrogen engines.

Gas Turbines

A detailed engineering trade-off study of LH_2's potential for providing improved gas-turbine efficiency and/or reliability is needed. Two parts are required in such a study:

(a) A parametric trade-off study of turbine cooling-air chilling: The parametric influences on the chilling of turbine cooling-air must be better established. Sizing and costing of equipment should be traded off with realistic analysis of performance gains over a range of operating variables.

(b) Study of heat-engine performance with LH_2 heat sink: same as in (a), except for heat engine.

Both of these subject areas can be viewed as on-board means for recovering the substantial enery previously "invested" as work of liquefaction, and hence, they are highly relevant to overall energy efficiency comparisons of hydrogen with other alternative fuels.

In the areas of development and demonstration of hydrogen as a fuel, efforts would certainly be required if any major commitment were to be made to large-scale hydrogen use. The necessary development areas are mentioned in the preceding Technical Results section. They include establishing long-term-reliability, state-of-the-art advances in H_2 gas-compression equipment and in LH_2 pumping and flow-control equipment. However, even before any long-term commitment to hydrogen use, open questions concerning these areas as well as safety and materials compatibility impede well-justified conclusions as to the practicality of H_2 relative to other fuel alternatives. A modest-scale, well-engineered demonstration could satisfy, at least partially, these concerns; the engineering development involved would necessarily force confrontation and resolution of with many of the key problems outlined herein. Experimental conversion and long-term operation of a small marine or stationary installation with a completely operational liquid-hydrogen fueling system is recommended.

SECTION 1

HYDROGEN-FUELED RECIPROCATING ENGINES

by

Richard B. Cole

Robert F. McAlevy, III

TABLE OF CONTENTS

SECTION 1 - HYDROGEN-FUELED RECIPROCATING ENGINES

LIST OF FIGURES

SECTION 1 - HYDROGEN-FUELED RECIPROCATING ENGINES

1.1 - BACKGROUND AND INTRODUCTION

A substantial body of experimental data has been accumulated for hydrogen-fueled reciprocating engines. Naturally-aspirated engines have been used for almost all of this work, despite the performance and operational shortcomings that they exhibit when operated on H_2 rather than gasoline. As early as 1933 (Ref. 1), however, it was demonstrated that certain of these shortcomings could be eliminated simply by injecting the H_2 directly into the cylinder following closure of the air intake valve. Despite the potential of this mode of operation, it appears that only a few other H_2-fueled engines have been operated in this way (Ref's. 2,3, 4,5).

H_2/O_2 engines were developed in the 1950's and 1960's (Ref's. 6&7) as "packaged" systems for space applications and more recently with the intention of radically reducing pollutant emissions (Ref. 8).

H_2 has also been partially substituted for gasoline in order to allow leaner operation than is possible with naturally-aspirated engines fueled by only gasoline.* Engines have been operated on this "mixed" fuel (Ref's. 9&10).

Compression-ignition engines have not apparently been operated successfully with pure H_2 as a fuel. For example, reliable compression ignition has not been achieved in tests at compression ratios as high as 29/1 (Ref. 11). It is apparently necessary to inject another fuel for reliable, controllable "pilot" ignition of the H_2 (Ref. 12). H_2 engine data for high compression ratios have typically been obtained with spark-ignition or glow-plug ignition.

During the present program, performance data for many of the H_2-fueled engines have been assembled, evaluated and reported a year ago (Ref. 13). Since then: (i) additional data have been obtained and evaluated: (ii) the theoretical performance of H_2-fueled

*conventional Otto-cycle with throttling engines has been calculated using the fuel/air-cycle approximation; (iii) the influence of cylinder fuel injection on engine cycle efficiency has been analyzed using the air-standard-cycle approximation.

H_2-fueled engine performance is compared frequently herein to gasoline-fueled engine performance. Thus, the similarities and differences between the relatively unknown entity and the universally known entity can be readily drawn. This approach should make it easier to perform "trade-offs" in evaluating H_2 as an alternative fuel for the future.

The new work reported herein, in conjunction with the previous work, represents a rational basis for the evaluation of H_2 as a reciprocating-engine fuel. But due to shortcomings and incompleteness in existent H_2-fueled engine data, the results of the subject evaluation must be considered tentative.

Most importantly, a complete experimental "performance map" of H_2-fueled engine data does not exist, thus undermining the confidence level of comparisons and conclusions based on the fragmentary available data collected and evaluated herein.

More generally, almost all of the H_2-fueled reciprocating engine performance data have been obtained with naturally-aspirated, air-breathing, spark-ignition, Otto-cycle engines. All such engines have been designed and in varying degrees optimized (over decades) for use with gasoline as a fuel. Typically, only minor modifications were made in adapting these engines for H_2 operation; certainly they were not transformed into "hydrogen engines", optimized for the use of H_2. Thus, data from operations with H_2 as a fuel in a gasoline engine are at a congenital disadvantage *vis-a-vis* gasoline operation. Comparisons based on such data are intrinsically biased against H_2 as a fuel. However, there is no reasonable alternative, and therefore, such comparisons are made.

Finally, it appears that no reciprocating hydrogen-fueled engine has ever been operated on H_2 for more than a couple of hundred hours. This is certainly an inadequate period for evaluation of lubricant contamination and degradation by water, which could lead to high wear rates or failure. Consequently, practical engineering judgement concerning this aspect of hydrogen as a fuel for reciprocating engines cannot be prudently rendered at this time. It must await accumulation of suitable longevity data.

1.2 - PERFORMANCE CALCULATION FOR NATURALLY-ASPIRATED OTTO-CYCLE ENGINES AND EXPERIMENTAL RESULTS

1.2.1 H_2-Fueled Engines

The fuel/air-cycle approximation to the H_2-fueled Otto-cycle was used to calculate both indicated thermal efficiency (η_I) and indicated mean-effective-pressure (IMEP) as a function of engine compression ratio (CR) and fuel/air equivalence ratio, ϕ*. This level of approximation incorporates realistic thermodynamic properties of the fuel/air mixture and the combustion products, but idealizes the compression and expansion processes (assumed isentropic) and idealizes the combustion processes (assumed constant-volume and adiabatic). Details are presented elsewhere (Ref. 14).

Figure 1-1 is an updated version of Figure C.3-1 of Ref. 13 and shows η_I vs ϕ calculated for a compression ratio, CR, of 10/1. Also displayed is the envelope of hydrogen-fueled engine data from a variety of sources, all scaled to a common reference of CR=10/1. This was accomplished by multiplying the experimentally-determined value of η_I by the ratio of the air-standard-cycle efficiency at CR=10/1 to the air-standard-cycle efficiency at the experimental CR. The highest actual indicated efficiencies, as scaled, fall within 85 to 90% of those calculated for the H_2-fueled Otto-cycle, as long as the engines are operated at levels of ϕ that are not below about 0.4, i.e., as long as ϕ does not approach the fuel-lean limit of 0.15 or so.

Figure 1-2 is an updated version of Figure C.3-2 of Ref. 13 and displays IMEP vs ϕ calculated for CR=10/1. Also displayed are hydrogen-fueled engine data from a variety of sources, all scaled to a common reference CR(10/1) in the same way as was done previously

*ϕ is defined here as the actual fuel/air (mass) ratio divided by the stoichiometrically-correct ratio.

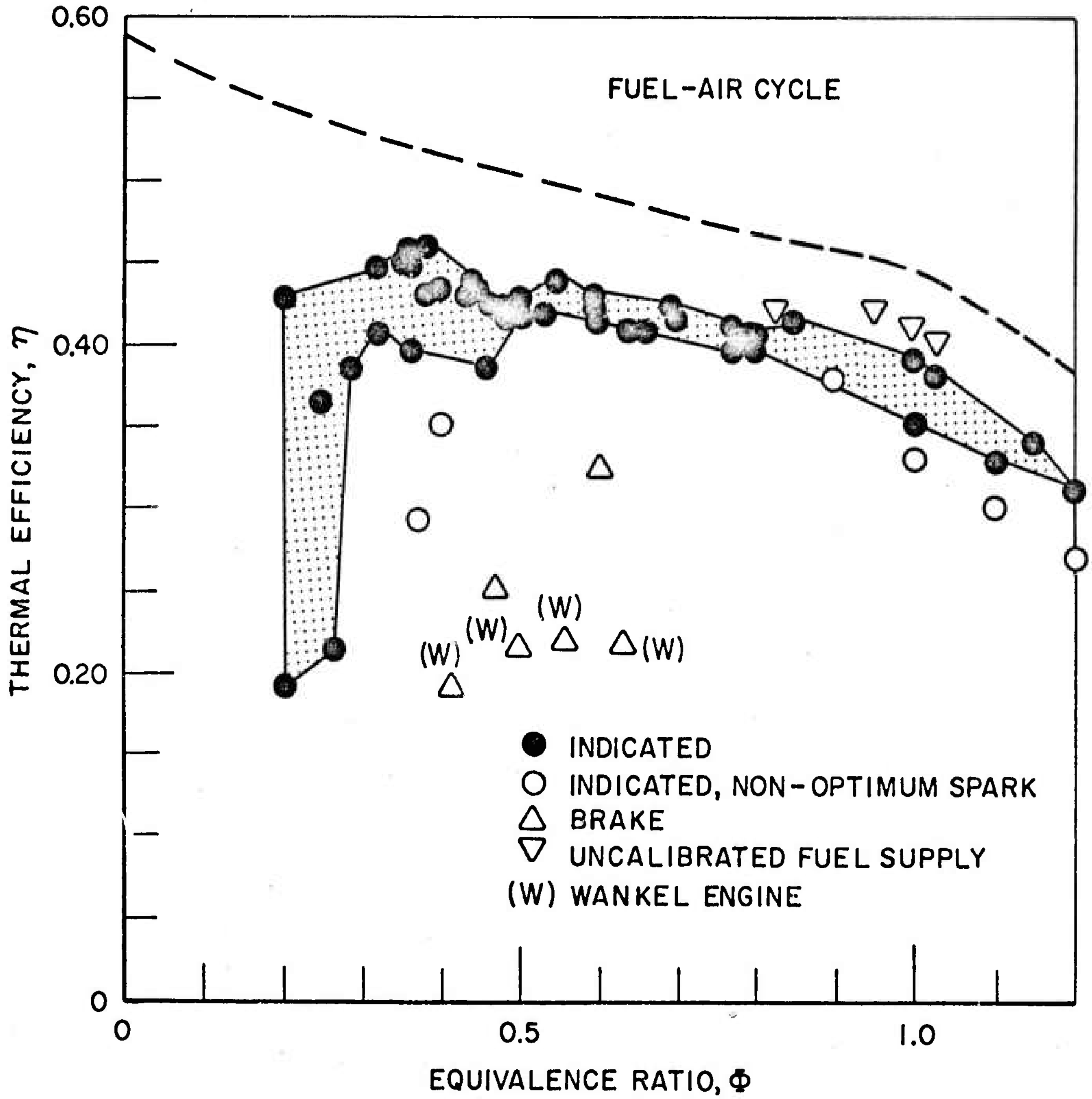

FIGURE 1-1: Collected Experimental Thermal Efficiencies of H_2/Air, Naturally Aspirated Engines (Data Scaled to Common CR = 10/1)

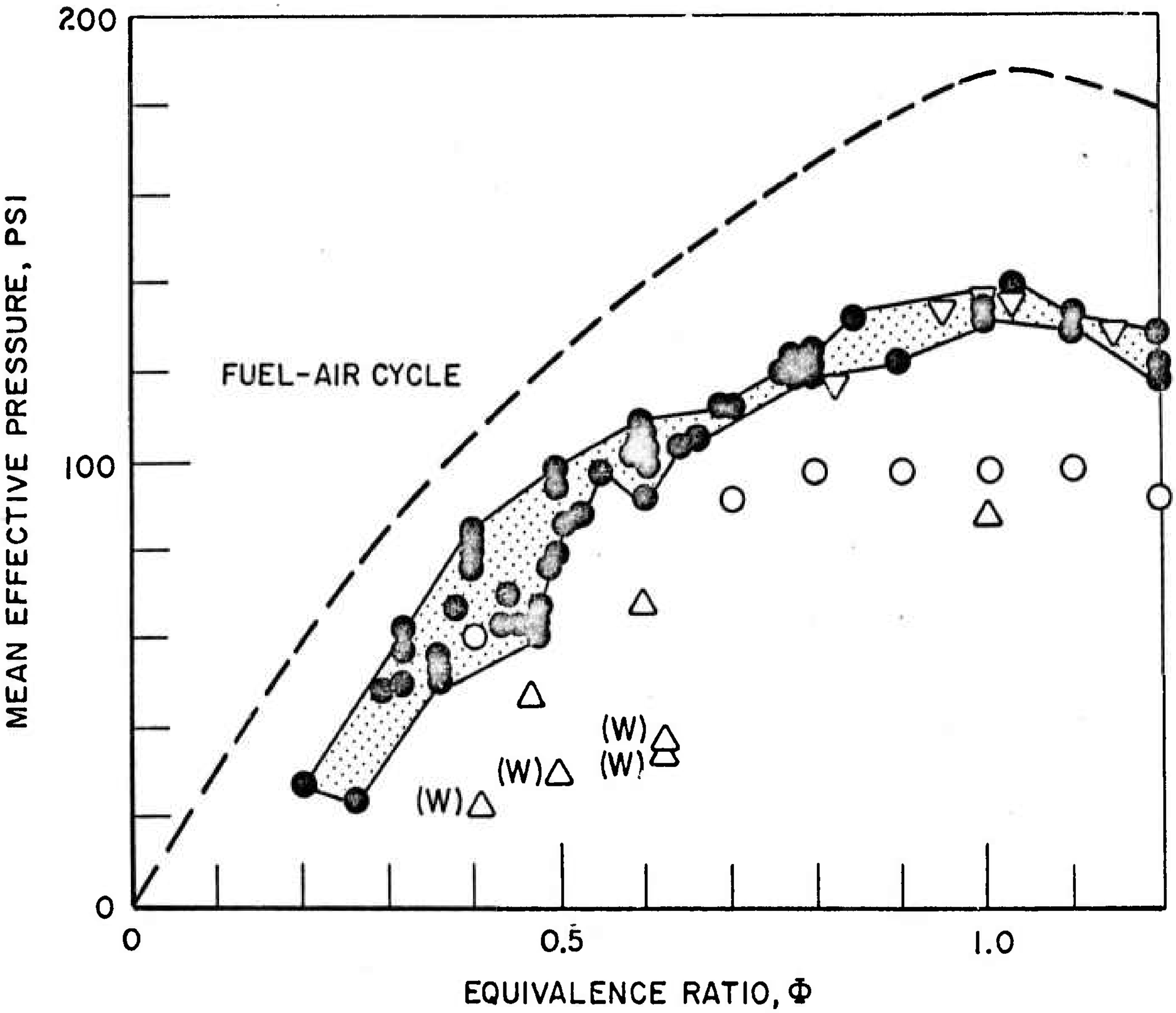

FIGURE 1-2: Collected Experimental Indicated Mean Effective Pressures of H_2/Air, Naturally Aspirated Engines (Data Scaled to Common CR = 10/1)

for efficiencies. The maximum actual IMEP values fall within about 70 to 80% of the calculated values.

The results of Fig's. 1-1 and 1-2 establish the validity of the calculated values from the fuel/air-cycle approximation as a basis for rationalizing the performance of H_2-fueled engines. As in the case of such comparisons with gasoline/air engines (Ref. 15), the actual and calculated values show the same trends and the actual values are low by a fairly constant factor, in a certain region of ϕ variation.

The differences between the calculated values of η_I and those obtained experimentally (Figure 1-1) result from the effects not modeled by the fuel/air-cycle approximation (e.g., heat loss, non-constant-volume combustion, finite time for valve opening and closing, etc.). For ϕ below 0.4 or so, these effects result in actual engine efficiencies falling below the 80 to 90% of calculated values which is characteristic of the region of ϕ>0.4. For ϕ below about 0.3 these effects can apparently result in a precipitous fall in actual engine efficiency below the calculated values; the data are quite scattered at low ϕ, and the confidence level of any conclusions based upon them is necessarily quite low. More investigation of this region of lean-mixture operation is needed.

The differences between the calculated values of IMEP and actual engine data (Figure 1-2) are due partially to the same effects that produce the shortfall in indicated efficiency. Additional influences also contribute to IMEP differences, such as pressure drop during induction of the fresh mixture into the cylinder, heating of the incoming mixture by hot engine parts, etc. These additional influences reduce the "volumetric efficiency" below that which is calculated within the context of the fuel/air cycle approximation. The fuel/air cycle approximation accounts only for the influence of residual combustion products on the incoming "fresh" charge. Comparison of the data with calculated values of IMEP yields values of volumetric efficiencies between 80 and 90%, comparable to or slightly higher than those exhibited by gasoline-fueled engines (Ref. 15).

The results of the fuel/air-cycle calculations can be conveniently displayed in the η_I-IMEP plane where, theoretically, lines of constant ϕ radiate from the origin. Figure 1-3 (Ref. 14) shows the results of fuel/air-cycle calculations over a wide range of ϕ and CR. Figure 1-4 displays the calculated results for CR=10/1 only and also the experimental data from Figures 1-1 and 1-2.

Figure 1-5 can be used to illustrate how the shortfall of actual engine performance below that calculated is due to (i) cycle nonidealities that majorly influence both η and IMEP (such as heat loss, nonconstant-volume combustion, etc.) and (ii) volumetric efficiency losses that influence only IMEP for the most part (pressure drop during charge induction, heating of the new charge, etc.). Cycle nonidealities result in displacement down from the calculated curve along lines of constant ϕ (e.g., for ϕ=1., from point (A) to point (B) in Figure 1-5). Losses in volumetric efficiency result in a reduction of IMEP along lines of constant η_I (e.g., from point (B) to point (C) in Figure 1-5). Thus, the actual data point at a particular value of ϕ is displaced to the left from the theoretically-calculated ϕ=constant lines radiating from the origin.

Differences between the volumetric efficiencies calculated for the fuel/air cycle and actual volumetric efficiencies can be obtained easily from such plots. For example, the ratio of the actual to the theoretical volumetric efficiencies at ϕ=1. is the ratio of IMEP at point (C) to that at point (B). Since theoretical volumetric efficiencies are very near unity (e.g., 0.98 to 1.01), these ratios are very nearly the actual volumetric efficiencies.

To summarize the preceding, the fuel/air cycle was used to calculate the performance characteristics of H_2-fueled reciprocating engines as has historically been common with gasoline/air engine. However, compared with gasoline-fueled engines, H_2-fueled engine performance characteristics are relatively scarce and have been obtained at a variety of disparate conditions with various engines.

The H_2-fueled engine data were scaled to a common compression ratio (10/1), and it was found that the upper limit of the data envelope was a relatively constant fraction of the calculated values

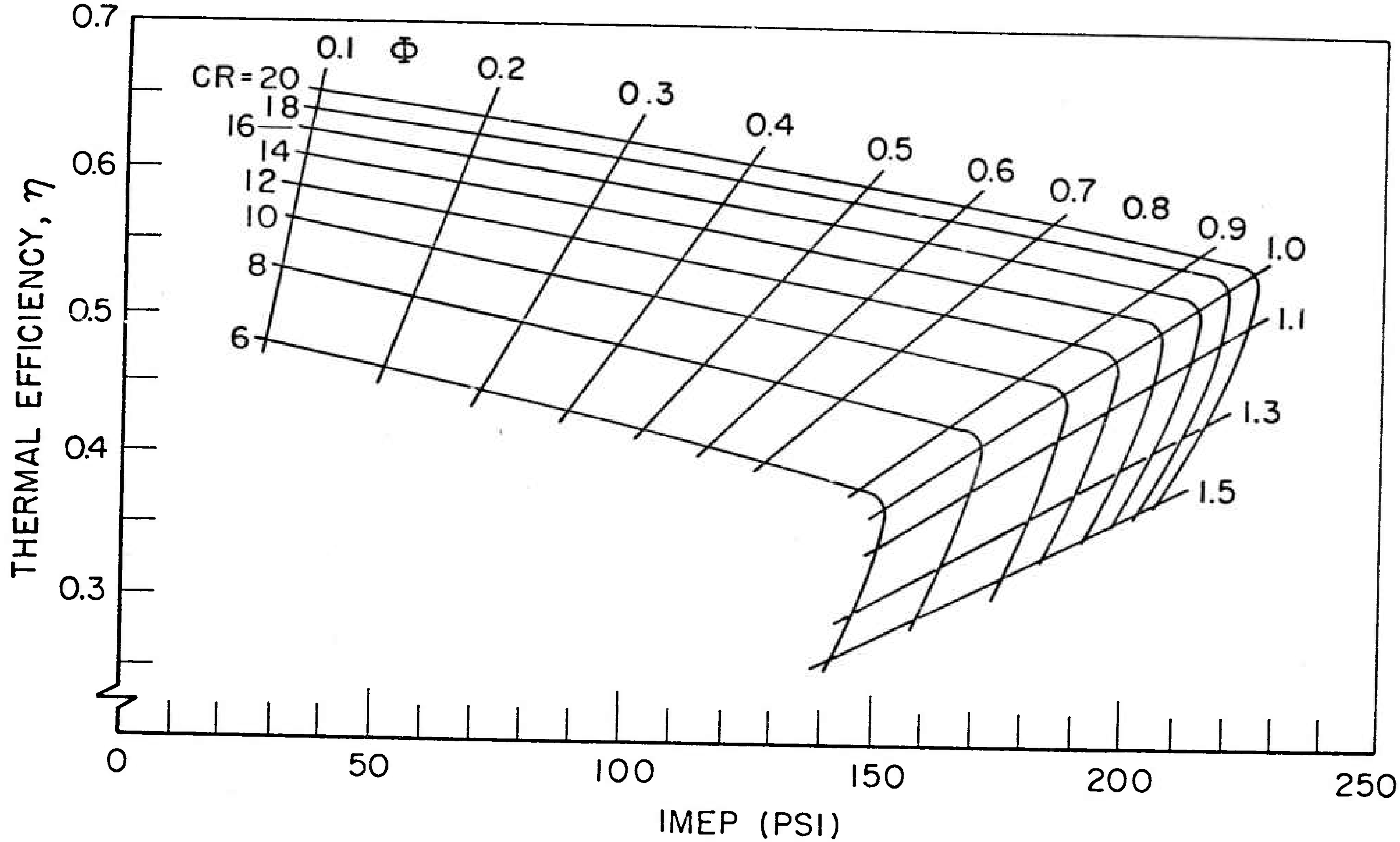

FIGURE 1-3: Performance Map for Naturally-Aspirated, Spark-Ignition, H_2/Air Engines ("Fuel-Air" Otto Cycle) (Ref. 14)

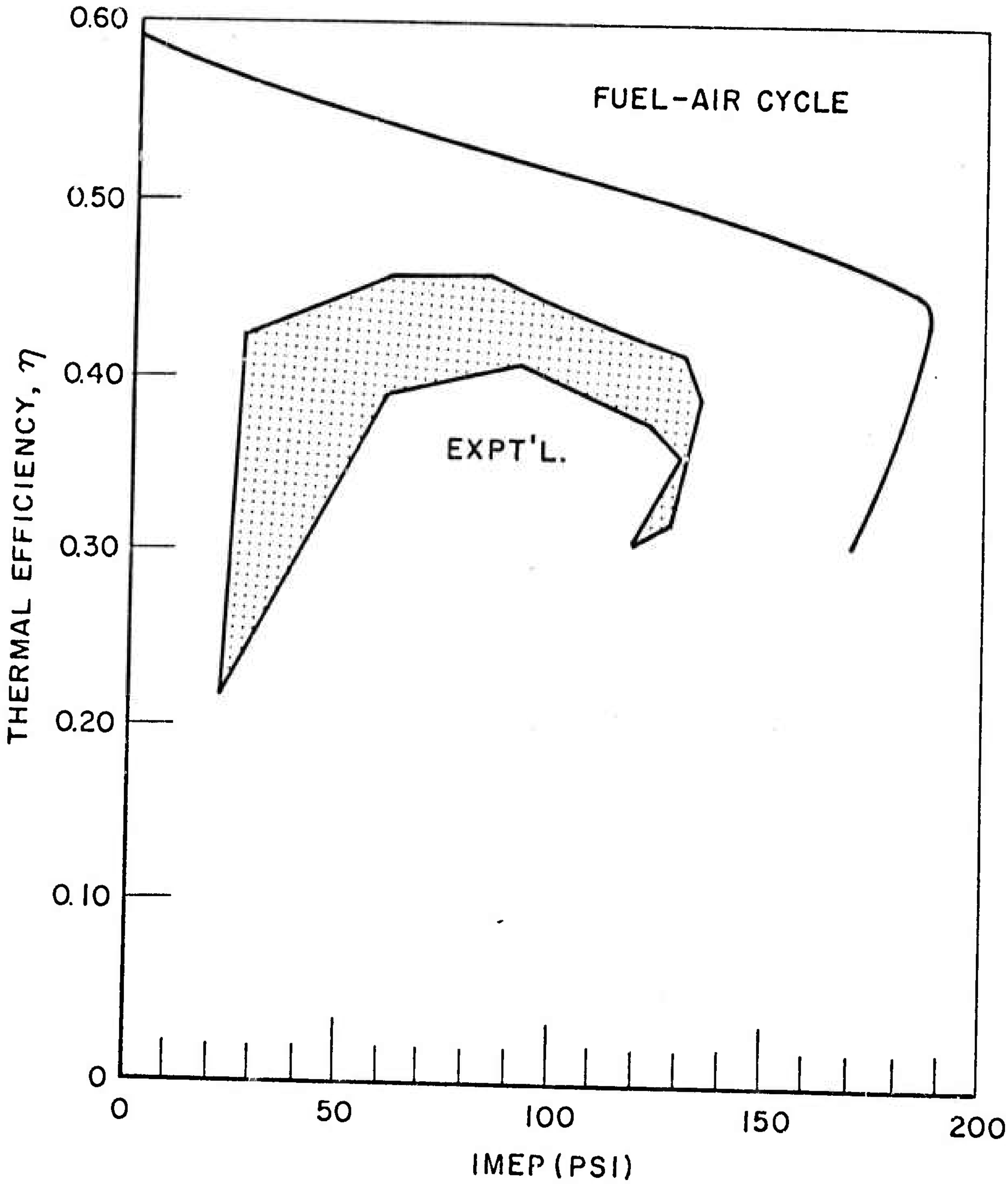

FIGURE 1-4: Theoretical and Experimental Performance Data for Naturally-Aspirated Otto Cycle (Experimental Data Scaled to Common CR = 10/1)

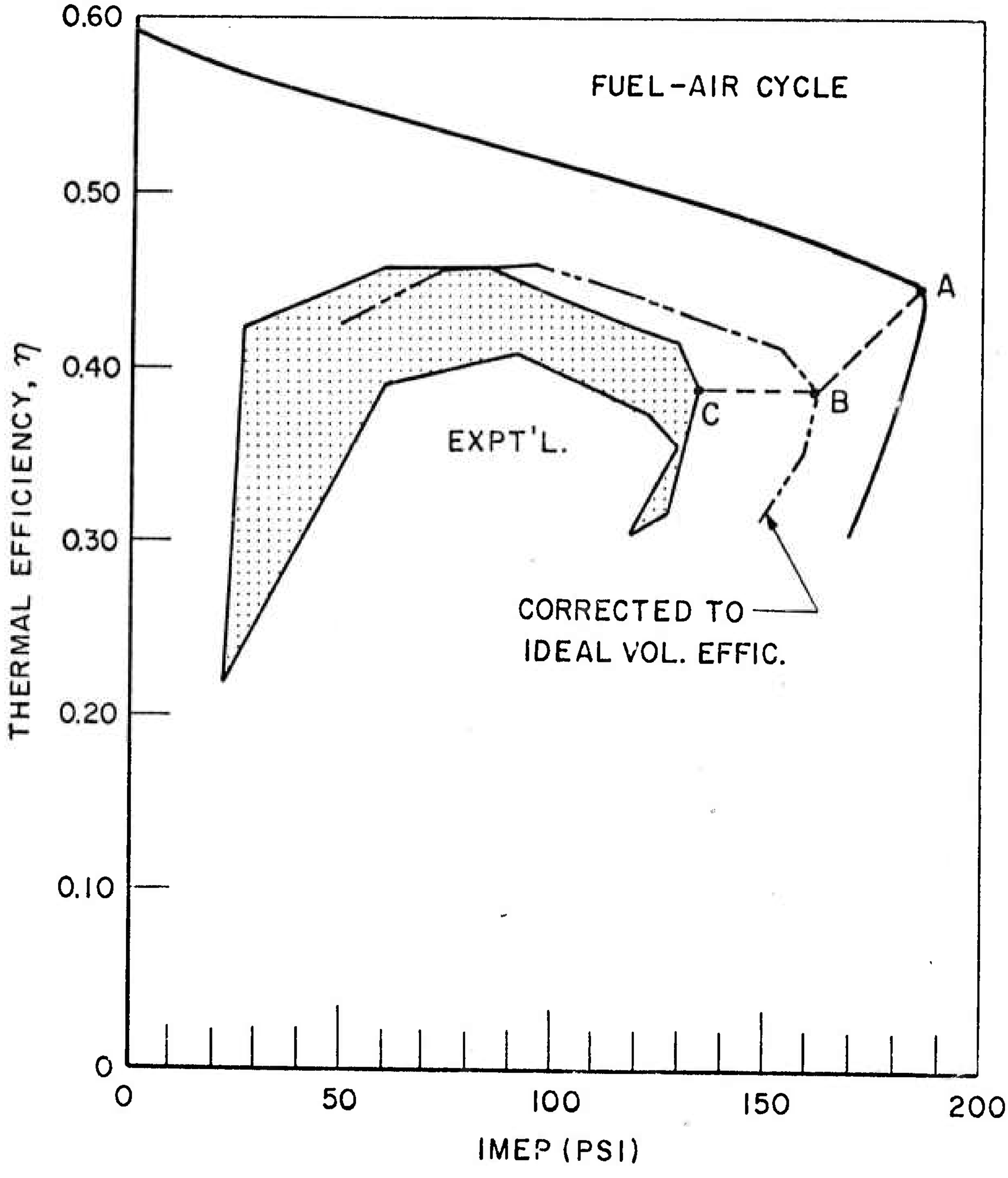

FIGURE 1-5: Graphical Representation of Volumetric-Efficiency and Cycle Non-Idealities (After Figure 1-4)

of η_I and IMEP in the ϕ region above 0.3 to 0.4. In this region, losses due to effects not modeled by the fuel/air cycle approximations are relatively small. It is concluded that the calculated values reasonably represent the performance of H_2-fueled engines in this ϕ region. Below ϕ=0.3 to 0.4, data are scant and too scattered to allow confident interpretation. More experimental data are needed in this lean-mixture region.

In the following sections, comparisons between H_2 and gasoline are made based on fuel/air-cycle calculations of η_I and IMEP while recognizing the validity of those for ϕ_{H_2} >0.3-0.4 and the uncertain relationship between calculated and actual values for ϕ<0.3. Conclusions are drawn with the expectation that comparisons based on future data will yield nearly identical results as comparisons based on the calculations in the ϕ region above 0.3 to 0.4.

1.2.2 Comparison of H_2 to Gasoline

1.2.2.1 Quality control of power output

The solid lines of Figure 1-6 show values of η_I vs IMEP at CR=10/1 for both H_2 (Ref. 14) and gasoline (Ref. 15) according to the fuel/air-cycle approximation. Both solid lines represent the case of "quality-control" of engine power output (i.e., variation of ϕ with "wide-open" throttle). Based on these calculations, gasoline is inherently superior to H_2 as a fuel as regards η_I (by several percent) and IMEP (by about 20% near $\phi \cong 1.0$, where maximum IMEP occurs). This fact is supported experimentally (Ref. 13) over the range of ϕ variation where operation on both fuels is possible.

In practice, pathological combustion-related phenomena, such as "knock", etc., prevent operation of unmodified, naturally-aspirated, H_2-fueled engines at maximum IMEP, e.g., above ϕ>0.65 or so with CR=10/1 (see Figure C.3-6 of Ref. 13). The actual engine efficiency falls noticeably from the calculated values for ϕ below 0.3 to 0.4, albeit the data are greatly scattered. Thus, operation at lower values of ϕ may be impractical in applications where fuel economy is important unless lean-mixture efficiencies can be improved. In practice, therefore, the range of ϕ variation in unmodified, naturally-aspirated, H_2-fueled engines will probably

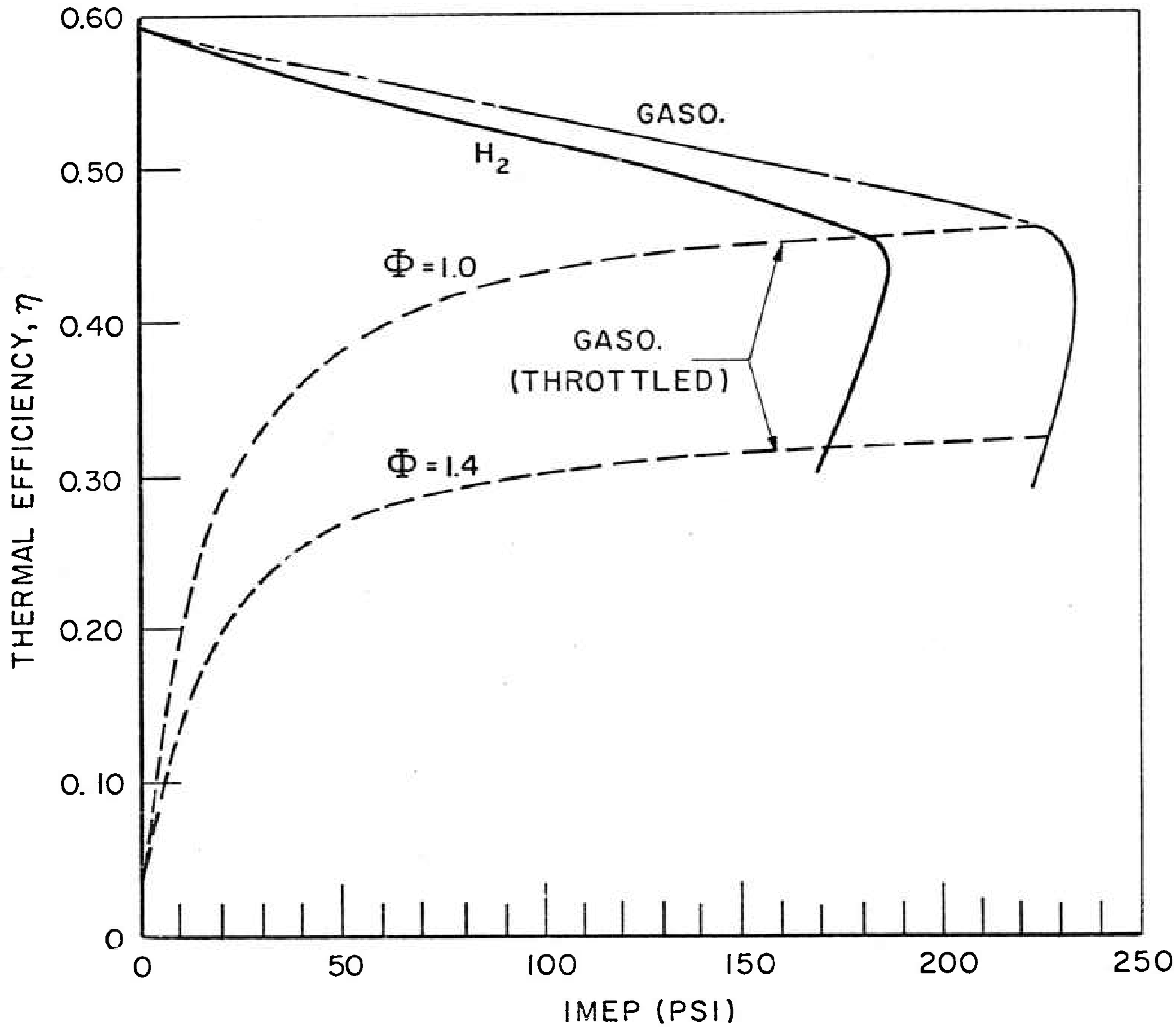

FIGURE 1-6: Theoretical Performance of H_2/Air and Gasoline/Air "Fuel-Air" Cycles (CR = 10/1)

be quite limited. In modified engines, the upper ϕ limit can be extended to beyond ϕ=1 (Ref. 13), but so far, only by accepting performance penalties that result from, for example, exhaust gas recirculation, nonoptimum spark-timing, etc.

In practice, gasoline-fueled, naturally-aspirated engines can operate at $\phi<1$ though the vast majority of current, production gasoline engines maintain ϕ near 1 and control power by throttles rather than quality control. However, lean-burn gasoline engines that operate normally at $\phi \cong .75$ are currently in production (Ref. 16) although they suffer from decreased "drivability". It might be possible to operate advanced types of lean-burn engines with quality control of power output in the near future. Presently, operation with gasoline at $\phi \lesssim 0.75$ requires either injection of "gasoline" directly into the cylinder, producing a "stratified-charge", or a two-chamber cylinder design. These can allow operation down to $\phi \cong 0.2$ (Ref. 17). Such gasoline engines allow quality control of power output. The theoretical performance of stratified-charge engines lies only slightly below the quality-controlled (homogeneous charge) line for gasoline in Figure 1-6 (Ref. 18, pg. 128), i.e., comparable in trend and values with those of H_2.

The scope of the present program did not permit gathering definitive data for stratified-charge gasoline-engine performance. However, based on calculated performance, it is expected that at comparable values of ϕ, gasoline in a stratified-charge engine can produce comparable η_I and higher IMEP than a H_2-fueled naturally-aspirated engine. There are indications that in the near future a large number of automobiles manufactured in the U.S.A. will employ stratified-charge or other type of lean-burn engines to take advantage of the efficiency and emissions benefits that accompany quality control of power output (Ref. 19).

1.2.2.2 Quality-controlled, H_2-fueled engines vs quantity-controlled gasoline-fueled engines

The power output of almost all currently used naturally-aspirated, gasoline engines is reduced by closing a throttle-valve on the mixture intake side. This reduces the density of the mixture entering the cylinder and, therefore, the amount of chemical energy available for conversion by the engine into mechanical work output. The process has been referred to as quantity-control of power output. It introduces "pumping losses" that result in decreasing cycle efficiency with decreasing power output. The dashed lines in Figure 1-6 were calculated for throttled gasoline-fueled engines operating at $\phi=1.0$ and $\phi=1.4$ (Ref. 14). At low loads, that is, low IMEP, the values of η_I calculated for quality-controlled H_2-fueled engines are seen to be much greater than those calculated for quantity-controlled gasoline-fueled engines.

As discussed at the end of Section 1.2.1, indicated performance data from H_2-fueled engines are scattered and fall precipitously below calculated valves in the region of ϕ below 0.3 to 0.4. However, it is in this region that the performance of quality-controlled, H_2-fueled engines is calculated to be superior to that of quantity-controlled, gasoline-fueled engines (compare the solid H_2-line with the dashed lines in Fig. 1-6). Thus, until more reliable H_2-fueled engine data are obtained in the region below $\phi\cong 0.4$, it will not be possible to determine if the indicated performance of quality-controlled, H_2-fueled engines are, in fact, superior to that of quantity-controlled gasoline-fueled engines - as shown by the fuel-air cycle calculations. For gasoline-fueled engines operating at $\phi\cong 1$, the efficiency data should be about 85% of the calculated valves (Ref. 15), whereas for H_2-fueled engines operating below $\phi\cong 0.3$ to 0.4 efficiencies start to fall precipitously from the calculated values (Section 1.2.1). The scope of the present program did not permit gathering the data required to actually carry out the comparison.

Friction losses reduce the indicated performance quantities to the values measured on the shaft - the "brake" quantities. A simplistic attempt has been made to account for friction within the context of the fuel/air-cycle-approximation calculations (Ref. 14). Compared with Figure 1-6, these results show that frictional effects strongly attenuate the calculated superiority in low-load performance of quality-controlled, H_2-fueled engines over quantity-controlled, gasoline-fueled engines. Again, the usefulness of these calculated results is undermined by the shortfall of H_2-fueled engine data below calculated valves in the region below $\phi \cong 0.3$ or 0.4 - exactly where H_2 is calculated to produce superior performance. Recently, however, some data have been published that support this superiority (Ref. 20).

1.3 - OXIDES OF NITROGEN GENERATED IN AND EMITTED FROM NATURALLY-ASPIRATED OTTO-CYCLE ENGINES

Limitations on the emission of oxides of nitrogen, NO_x, from automotive vehicles are imposed by Federal standards. Also limited are the emissions of carbon monoxide and unburned hydrocarbons. Hydrogen-fueled engines produce all three types of pollutants. The latter two originate from the engine lubricant and their emission from "unmodified" engines can exceed the Federal standards, under certain conditions (Ref. 9). The oxides of nitrogen originate from reactions between oxygen and nitrogen in the hot combustion products by means of a common mechanism in both H_2-fueled and gasoline-fueled engines.

In the cylinder of an engine operating on a homogeneous fuel/air mixture, soon after passage of the flame front, oxygen atoms react with nitrogen molecules to produce nitric oxide, NO. The rate of NO production, rate of NO destruction, rate of NO oxidation to higher oxides in the engine, and the NO_x concentration in the exhaust products at any stage of expansion during the power stroke, all depend on combustion-product oxygen concentration and temperature. Fuel properties unrelated to these two factors play negligibly small roles insofar as NO_x is concerned.

The concentration of NO, [NO], in the combustion product at piston top-dead-center is readily calculated within the framework of the fuel/air-cycle approximation (Ref's. 14,15). These are the values that would be reached, theoretically, given sufficient time for equilibrium to be established amongst the combustion-product gases.

Models allowing for the finite rate of chemical reaction, which also include the effects of finite rate of combustion and temperature gradients within the cylinder have been produced (Ref's. 10,22,23). They are referred to as "kinetic" models. The levels of NO_x existing at piston bottom-dead-center, and presumably in the exhaust emitted from H_2-fueled engines, have been calculated using these models.

1.3.1 Theoretical Predictions and Experimental Results

1.3.1.1 Engine operation in the region of $\phi>0.8$

Figure 1-7 reproduces the results of both the equilibrium, top-dead-center calculations of [NO] as well as the chemical-kinetics, bottom-dead-center calculations. In the region of $\phi>0.8$, the calculated exhaust-gas levels are only slightly lower than the top-dead-center equilibrium levels. This is because the chemical kinetic rate of NO_x destruction that actually takes place during expansion is extremely temperature dependent. Therefore, after only slight expansion during the power stroke the temperature falls to a level that does not permit further NO_x decrease.

Such "freezing" of NO at close to top-dead-center equilibrium levels has long been recognized in gasoline-fueled engine operation (Ref. 24). In fact, exhaust gas [NO_x] measured with gasoline-fueled engines have been found historically to be closely and conveniently correlated by the calculated top-dead-center [NO] equilibrium values. Measurements of NO_x in the exhaust gas from naturally-aspirated, H_2-fueled engines for $\phi<0.8$ operation (Ref's. 9,10) plotted in Figure 1-9 are also well correlated on the basis of top-dead-center equilibrium values of [NO].

1.3.1.2 Engine operation in the region of $\phi<0.8$

The combustion-product temperature decreases with decreasing ϕ, and when ϕ falls below 0.8 or so the temperature-sensitive rate of NO generation cannot produce the levels that are demanded in order to reach equilibrium [NO]. This results in the extreme divergence between the predictions of the kinetics models and the prediction of the equilibrium model with falling levels of ϕ as can be seen in Figure 1-7. Experimental data are, however, correlated well for $\phi<0.8$ with the results of the chemical-kinetics-model calculations. Therefore, in contrast to the situation in the region of $\phi>0.8$, the equilibrium model is inadequate in the region $\phi<0.8$, or so, and its results are useless there for correlation of NO_x emissions. The more complex chemical-kinetics model must be used for $\phi<0.8$.

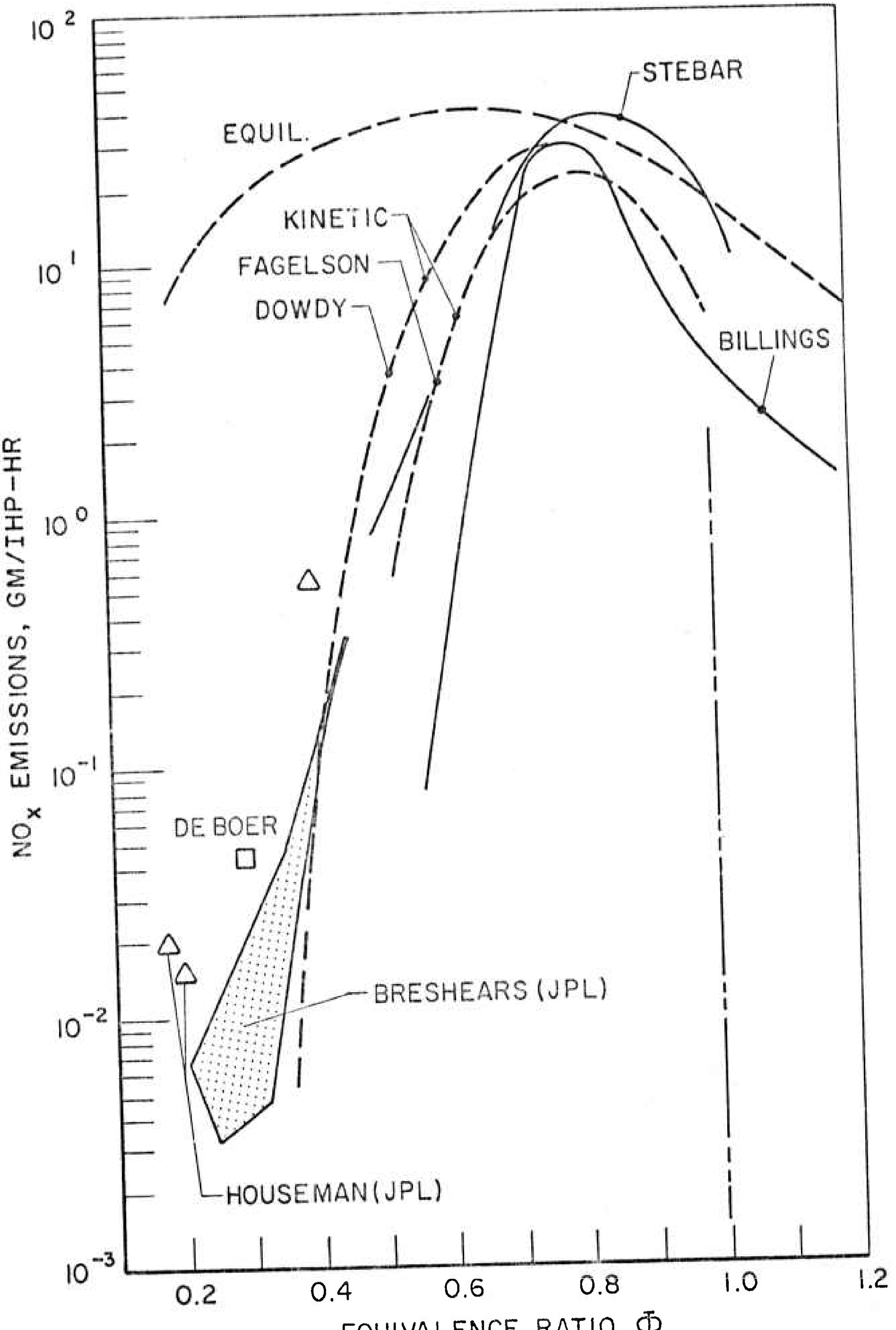

FIGURE 1-7: Theoretical and Experimental NO_x Emissions From Naturally-Aspirated H_2 /Air Engines (Various CR's)

1.3.1.3 Comparison of H_2 to gasoline

The combustion-product temperature is greater, by about 6%, in H_2-fueled engines than in gasoline fueled engines, when operated on homogeneous fuel/air mixtures at the same value of ϕ. Thus, at any value of ϕ the amount of NO_x generated and emitted for gasoline-fueled operation (Ref's. 9,10,21,22) could be expected to be less than that calculated for H_2-fueled operation (Fig. 1-9). Experimentally, engine factors such as spark timing have a powerful influence on NO_x emissions. However, in cases where both fuels have been used to operate the same engine, the NO_x emissions with H_2 operation were found to be greater by a factor of 5 or so (Ref's. 10,25) or at least a factor of 2 or so (Ref's. 9,26), compared to NO_x emissions with gasoline operation at the same value of ϕ.

So, in theory and in practice, when compared at the same value of ϕ, reciprocating engines operating on homogeneous fuel/air mixtures show greater NO_x generation and emission with H_2 than with gasoline. It can be assumed that "fixes" to reduce NO_x emissions, such as exhaust gas recirculation, "lean-burn", introduction of water into the unburned mixture, upsetting the spark timing for maximum power output, etc. will all have to be more extreme for H_2 operation than with gasoline operation. Detailed investigation of this likelihood was not permitted within the priorities of the subject program.

The preceding concern for NO_x has involved only engines operating on homogeneous fuel/air mixtures. However, as discussed in Section 1.2.2.1, gasoline engines with homogeneous fuel/air mixtures do not normally operate below $\phi \cong 0.75$ even as "lean-burn" engines. To operate on mixtures that have an overall $\phi \ll 1$, it is necessary to use a stratified charge, with a large spatial variation of local ϕ throughout the cylinder. Production of NO_x responds to local combustion conditions, and the actual level of NO_x emission from such engines should be many times greater than the levels measured when homogeneous lean fuel/air mixtures are burned (Fig. 1-7). Time did not permit gathering of NO_x emissions data from gasoline-fueled, stratified-charge engines. Nor was a review made of theoretical predictions of NO_x emissions from such engines within the priorities of the subject program. It might be expected, however, that at a given value of ϕ, a gasoline-fueled engine burning a stratified charge would emit substantially greater quantities of NO_x than a naturally-aspirated, H_2-fueled engine burning a homogeneous fuel/air mixture.

1.4 - HYDROGEN "SUPERCHARGED" ENGINES: DIRECT CYLINDER FUEL INJECTION FOLLOWING CLOSURE OF THE AIR INTAKE VALVE

The power output of naturally-aspirated air-breathing engines is limited by their ability to ingest air for oxidation of the fuel supplied. During engine operation, air does not fill the entire displacement volume of the cylinders due to the presence of combustion product gases remaining from the previous cycle, moisture in the ingested air and the presence of the fuel itself. At ϕ=1, gasoline vapors occupy about 2% of the displacement volume while H_2 occupies about 30%. This suggests, for example, that approximately a 30% increase in the number of moles of H_2/air mixture at ϕ=1 can be achieved by: (1) ingesting only air during the intake stroke ,then; (2) injecting H_2 directly into the cylinder following closure of the air intake value. Such operation is referred to as "CFI" for direct cylinder fuel injection following closure of the air intake valve.

In addition to allowing increased mean effective pressure (increasing the work per cycle per unit of engine displacement), CFI operation holds the potential of eliminating "backfire", "preignition" and other of the pathological engine-combustion phenomena (Ref. 13) that plague the operation of H_2-fueled, naturally-aspirated engines.

Apparently, Rudolf Erren (Ref. 1) first suggested CFI operation of H_2-fueled reciprocating engines and claimed superior engine performance. For example, Erren claimed (Ref.1) "one could get nearly twice as much power with hydrogen as with petrol". But quantitative data to support this claim were apparently never published or validated independently.

In this section, the fundamental basis for evaluating H_2-fueled, CFI Otto-cycle engine performance is developed. Published data obtained with several different engines are reviewed against this background. Along with one additional effort (Ref. 5) from which data are not available in useful form, these engines have produced all the CFI data known to the authors. These data were all obtained on single-cylinder, laboratory engines using compressed-gas bottles as a fuel reservoir.

1.4.1 Calculated C.F.I. Engine Performance

Section 1.2.2 presented calculated results that showed gasoline to be inherently superior to H_2 as a naturally-aspirated, reciprocating-engine fuel. See Figure 1-6, for example. For naturally-aspirated engines, the η_I for gasoline is greater by several percent and the maximum IMEP is greater by about 20% at $\phi \simeq 1$, for example.

The reasons for gasoline's superior IMEP is explained in Ref. 14: The 20% decrease in maximum IMEP with hydrogen arises from three major sources. First, at any given inlet pressure and temperature the maximum volumetric heating value of hydrogen/air mixtures is appreciably lower (about 15%) than with gasoline/air mixtures. Second, a stoichiometric hydrogen/air mixture theoretically undergoes about a 15% decrease in the total number of moles upon burning compared with about a 7% increase with gasoline/air; pressures throughout the engine cycle tend, therefore, to be reduced with hydrogen. Lastly (and to a modest extent counter-balancing the first two factors), maximum hydrogen/air flame temperature is about 6% higher than the corresponding gasoline-air value; this acts to increase pressures throughout the engine cycle. Along with some lesser sources of differences, these yield the net decrease of about 20% in work per cycle per unit engine displacement, IMEP.

Of these factors, only the volumetric heating value will be significantly affected by converting from naturally-aspirated to CFI operation. That is, by increasing the quantity of mixture in the cylinder, while holding ϕ constant, the volumetric heating content and thus IMEP will be increased. But, as discussed in 1.4.1.1, η_I will be relatively unaffected by converting from naturally-aspirated to CFI operation.

The performance characteristics of H_2-fueled CFI engine operation were derived approximately from the results of fuel/air-cycle calculations made for naturally-aspirated engines (Fig. 1-6). For CR=10/1, the values of IMEP at different values of ϕ as shown in Fig. 1-6 were multiplied by the ratio of volumetric heating value for CFI operation to that of naturally-aspirated operation, and were plotted at the same value of η_I in Figure 1-8 as the line labeled "H_2 CFI".

At ϕ=1, the ratio of IMEP for H_2-fueled CFI operation (254 psi from Fig. 1-8) to IMEP for gasoline-fueled operation (237 psi from Fig. 1-6) is about 1.07. Thus, for engines of the same total displacement and speed, the power output from a H_2-fueled CFI engine should be about 7% greater than that from a gasoline-fueled naturally-aspirated engine. This example calculation does not support the claim of Erren (Ref. 1) that the power output should be nearly 100% greater.

1.4.1.1 Air-standard-cycle analysis of the gaseous-fuel-injected supercharged Otto-cycle

Air-standard-cycle analysis can be made without the aid of a computer since the thermodynamic properties of the working fluid are all assumed constant and equal to that of air. The results are less realistic than those calculated using the fuel/air-cycle approximation. Nevertheless, they are useful in predicting overall cycle behavior and for examining the differences between cycles, i.e., the unmodified Otto-cycle vis-a-vis the CFI Otto-cycle. In Appendix 1-A, the CFI Otto-cycle is analysed on the basis of the air-standard-cycle approximation, and the results are reproduced here.

For engines supplied with H_2 from pressurized bottles, as they were in the cases for which data have been published (Ref's. 3,4,5), the analysis yields:

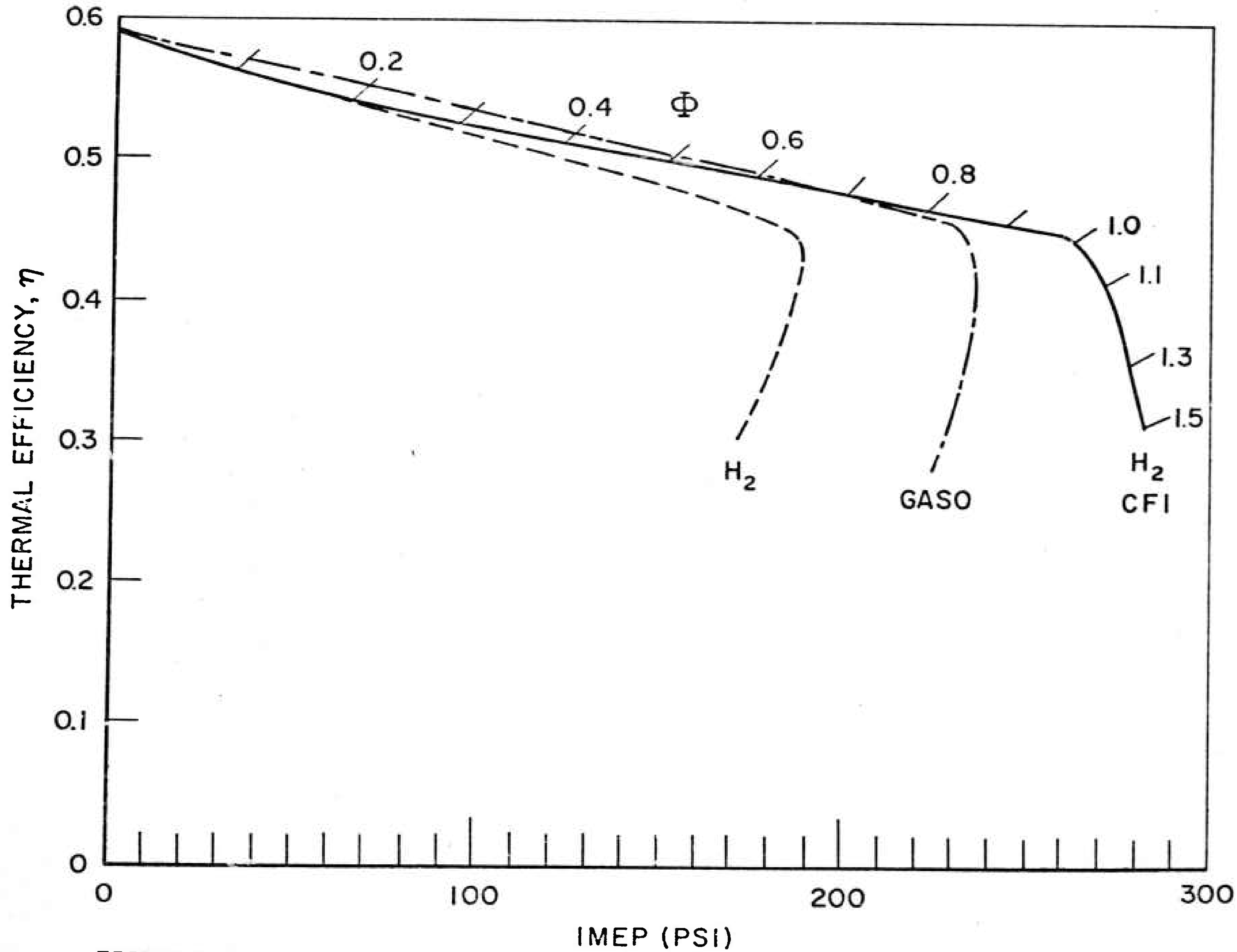

FIGURE 1-8: Approximate Theoretical Performance of H_2/Air Engine with Cylinder Fuel Injection (CFI) and CR = 10/1 [after Fig. 1-6]

$$\eta_{I,CFI} = \eta_I + \left(1 - \frac{1}{(V_1/V_x)^{\gamma-1}}\right) \frac{C_P T_F}{\Psi} \qquad (1)$$

where:

$\eta_{I,CFI}$ ≡ CFI Otto-cycle indicated efficiency based on net piston work

η_I ≡ Naturally-aspirated Otto-cycle indicated efficiency based on net piston work

V_x ≡ Cylinder volume when fuel injected

V_1 ≡ Cylinder volume at piston bottom-dead-center

C ≡ Fuel molar specific heat at constant pressure

Ψ ≡ Fuel molar chemical energy

T_F ≡ Fuel reservoir absolute temperature

For the case of a "gratuitous" high-pressure fuel supply, the efficiency of the CFI cycle is greater than that of the unmodified Otto-cycle if $V_x > V_1$, i.e., as long as the piston has moved from its bottom-dead-center position before the cylinder is charged with fuel. Then an increment of work can be derived by expansion of the injected fuel mass, having enthalpy $C_P T_F$, through a compression ratio of V_1/V_x. The efficiency benefit from this increment of work is seen in Eq. 1.

The maximum increment of work is derived for fuel injection at piston top-dead-center. Then,

$$\frac{\eta_{I,CFI}}{\eta_I} = 1 + \frac{C_P T_F}{\Psi} \qquad (2)$$

For injected H_2 as hot as 1,000°R, $C_P T_F/\Psi = 0.066$, implying a 6.6% increase in engine-cycle efficiency due to injection at elevated pressure. For injection at room temperature (560°R), $C_P T_F/\Psi = 0.037$. Thus, if the H_2 reservoir temperature is not unrealistically high, $\eta_{I,CFI} \cong \eta_I$, and the basic assumption underlying construction of Figure 1-10 from Figure 1-6 is valid. Assuming $\eta_I \simeq 0.30$, then $\eta_{I,CFI}$ should be no more than a percentage point or so greater in laboratory tests of CFI-operated, H_2-fueled engines when the fuel is supplied gratuitously from room-temperature gas cylinders.

Should it prove necessary to compress gaseous hydrogen for high-pressure fuel injection, engine power would presumably be used. The consequent efficiency loss (see Appendix 1-A) is likely to be greater than the small gain anticipated with CFI operation. In such a case, further attention to alternative schemes for H_2-compression appears justified.

1.4.2 Experimental Performance Characteristics of H_2-Fueled CFI (Single-Cylinder) Laboratory Engines

It appears that only a few H_2-fueled engines have been successfully modified for CFI operation. They all were single-cylinder, laboratory engines. In each case, the H_2 was supplied "gratuitously" to the engine from laboratory gas cylinders, i.e., the work of compression for the high-pressure fuel gas is not charged against the engine power output or efficiency. There were no reports of "backfire" phenomena, and none should be expected since H_2 was not introduced to the cylinder until the air intake valve was closed and the compression stroke was started. "Rough running" at high ϕ and knock at high ϕ and CR have, however, been reported (Ref's. 4,5).

At typical engine speeds, only a few milliseconds are available between the time that fuel injection is initiated, during the piston compression stroke, and the time that further combustion of the fuel already injected becomes inefficient, during the piston power stroke. Thus, in order to achieve good performance with a CFI engine it is necessary for the injection system to achieve prompt mixing of the injected H_2 with the air contained in the cylinder.

Oehmichen (Ref. 4) developed an injector system which introduced H_2 into the cylinder during the first half of the compression stroke from an injector located in the sidewall and operated at a pressure of about 6 atm. The best performance was obtained by directing the H_2 jet tangential to the cylinder wall and by carefully controlling the cooling-water temperature.

In order to approximate the effect of CFI on Oehmichen's reported power outputs and efficiencies, Eq. 1 was evaluated. To evaluate V_1/V_x in Eq. 1, it was assumed that all of the H_2 was injected when the pressure in the cylinder reached 6 atm., and that the air contained in the cylinder had been compressed isentropically from 1 atm. (at piston bottom-dead-center, i.e., V_1) to 6 atm. (at a volume equal to V_x). Thus, $V_1/F_x = (6/1)^{1/\gamma} =$ 4.28, assuming $\gamma = 1.4$. Assuming further that the H_2 supply was at a room temperature, then Eq. 1 predicts that the indicated efficiency of his engine should be greater by about 1.5% than the efficiency of a naturally-aspirated engine of the same compression ratio.

The data of Oehmichen at CR=10/1 and an engine speed of 1500 rpm are plotted in Fig. 1-9. In contrast with the "theoretical" curve of Fig. 1-9, Oehmichen's data are limited to mixtures leaner than stoichiometric ($\phi \leq 0.6$) and consequently do not show IMEP's greater than about 180 psi. Oehmichen did not, owing to high rates of pressure rise ("knock"), operate at stoichiometric mixture ratios.

For $\phi<0.4$, the Oehmichen data exhibit values that are within about 90% of the "theoretical" curve for a given value of ϕ, indicating that the fuel-injection system performed well. For $\phi<0.4$, the data trend moved more sharply away from the theoretical curve than is apparently the case with naturally aspirated engines (Fig. 1-4). This suggests that losses in Oehmichen's CFI engine became relatively greater than those exhibited by naturally-aspirated engines as the fuel-lean limit was approached ($\phi<0.4$).

Near and above $\phi=0.4$, however, the reverse may be true. That is, the losses from the best-performing naturally aspirated engines appear to be greater than those from Oehmichen's CFI engine. For example, at $\phi=0.4$ and CR=12/1 Oehmichen reported an indicated efficiency of 51% when his engine was operated at 1500 rpm. King

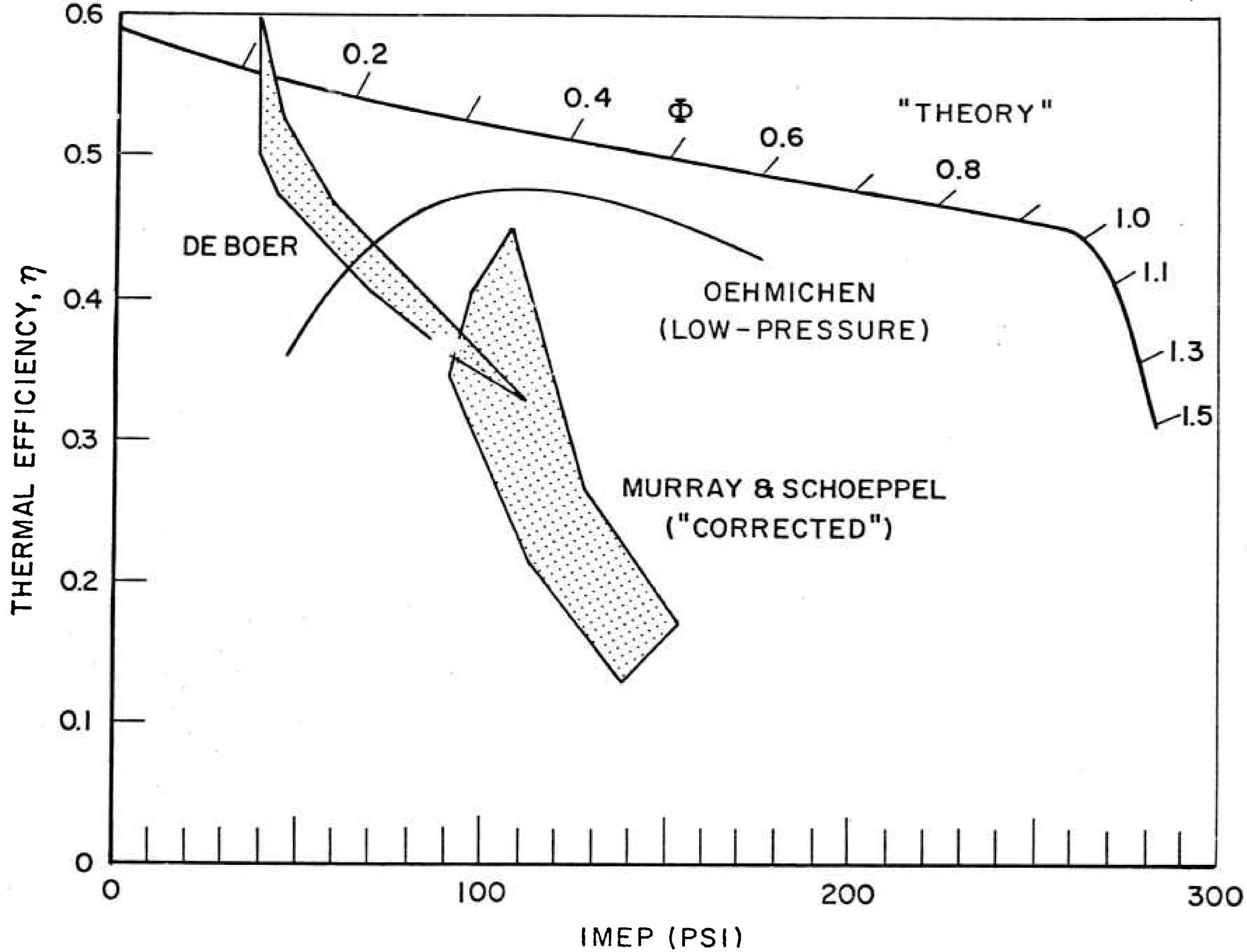

FIGURE 1-9: Theoretical and Experimental Performance Data for CFI H_2/Air Engines (Experimental Data Scaled to Common CR = 10/1)

(Ref. 27), who appears to have worked more carefully than other investigators of naturally-aspirated engines, has achieved the highest reported efficiencies with them. At the conditions cited, however, King reported an indicated efficiency of only 47%. Even when Oehmichen's efficiency value is divided by 1.015 (to account for gratuitous supercharging work), it still is 3 percentage points greater than King's value from a naturally aspirated engine.

Murray and Schoeppel (Ref. 3) used a small (3.5 hp, CR=6.5/1) engine to generate brake (i.e., not indicated) performance data. Fuel was introduced very late in the compression stroke and injection continued past the time of spark-ignition and the point of peak cylinder pressure was achieved. On occasion, injection was not terminated until the piston was well into the expansion stroke.

The data of Murray and Schoeppel were scaled to CR=10/1 using the technique outlined in Section 1.2. In order to plot indicated as well as brake quantities, in Fig. 1-9, it was necessary to deal with the mechanical efficiency of Murray and Schoeppel's engine, a quantity which was not measured by them. For this purpose, a reasonable friction mean-effective-pressure was assumed (30 psi at 3500 rpm). In addition, 1-hp additional indicated power output was credited to the engine, based on Murray and Schoeppel's estimate of the power required to drive their experimental injector. Even with such corrections, Murray and Schoeppel's data (Fig. 1-9) indicate low efficiencies relative to both Oehmichen's data and to theory except at the lowest loads (ϕ's) tested. Figure 1-10 emphasizes this fact on different coordinates.

Murray and Schoeppel's corrected data are seen in Fig. 1-10 to fall short of Oehmichen's indicated values. It is concluded that Murray and Schoeppel's fuel/injection system probably did not perform well. This suggests also that the very low NO_x emissions reported by Murray and Schoeppel may derive from inefficient injection and combustion rather than representing a fundamental benefit of their injection scheme (or of hydrogen as a fuel).

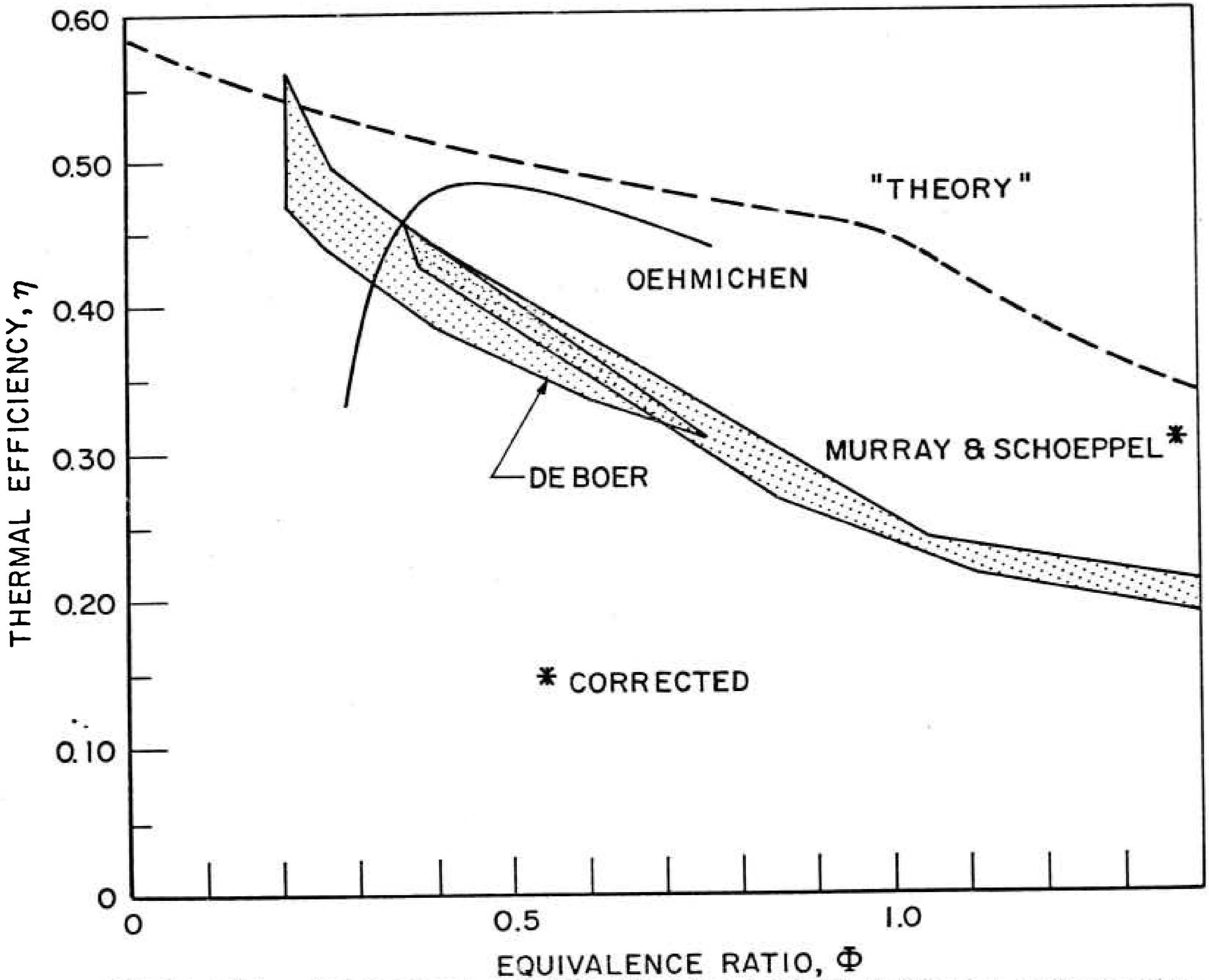

FIGURE 1-10: Theoretical and Experimental Indicated Thermal Efficiencies for CFI H_2/Air Engines (CR = 10/1)

de Boer, et al (Ref. 5) used a modified CFR single-cylinder laboratory research engine. The fuel injector was located at the top of the cylinder and the H_2 jet was directed radially inward. Unlike the injection timing of Murray and Schoeppel's engine, fuel injection was terminated prior to spark-ignition. From Eq. 1, and assuming that $V_x/V_1 = 5/1$ the efficiency of the engine should be less than a percentage point greater than a naturally-aspirated engine operating at the same compression ratio. The envelope of De Boer's most recent published data, for early injection and a relatively long injection time, are plotted in Fig. 1-8. Near the fuel-lean limit, they are comparable to that of Oehmichen and the "theory" curve. But as ϕ is increased, they fall sharply from both, indicating much greater losses than were evidenced by the Oehmichen engine. It is concluded that the fuel-injection system employed did not work well.

From the limited data available concerning CFI operation, it appears that, except for Oehmichen's efforts, CFI engines have demonstrated low efficiencies and mean effective pressures relative to theory except at very low power output (e.g., $\phi \leq 0.4$). Oehmichen's results, however, exhibit high efficiencies relative to the ideal theoretical cycle but only up to modest power (IMEP) level ($\phi \cong 0.6$). Figure 1-9 indicates that the performance region for which the CFI engine conceptually shows promise, that of high IMEP, has only slightly been touched experimentally; Oehmichen's reported IMEP's are, at a given compression ratio, higher than those reported from any of the diverse naturally-aspirated hydrogen engines. However, Oehmichen's improved IMEP with CFI are not greater by the margin which should be possible at richer mixtures than he was able to test successfully. This is probably because more injector development is required, a fact which was given substantial attention over 40 years ago by Rudolf Erren. Unfortunately, Erren's efforts and their results are not apparently well documented.

In summary, some experimental data are available which partially document the potential of CFI operation for improving hydrogen-engine performance beyond that possible with natural aspiration.

The most promising hydrogen reciprocating engine presently apparent is a cylinder-fuel-injected engine of elevated compression ratio ($\leq 10/1$) with provision for charge stratification and/or other means of combustion-control to reduce rates-of-pressure-rise to manageable levels (even under near-stoichiometric conditions). The development of improved, practical fuel-injection schemes is a critical problem for such an engine. As well as providing low rates-of-pressure-vise during combustion, the injection system should provide for minimum NO_x formation. This last desirable feature is by no means assured by prior experimental (or analytical) efforts with CFI hydrogen engines.

1.5 - MIXED-FUEL HYDROGEN ENGINES

A previous report (Ref. 13) reviewed earlier efforts toward developing and characterizing engine operation with hydrogen as a fuel in combination with other liquid and gaseous fuels. Since the time of that report, additional work has been done at Jet Propulsion Laboratory concerning the addition of hydrogen to gasoline. The continuing intent of the JPL program is to extend the lean limits of combustion of gasoline by the addition of hydrogen generated from gasoline.

As noted earlier (Ref. 13) and corroborated by more recent data (Ref. 10), engine efficiencies for such mixed fuel operation are comparable with or slightly higher than those for quality-controlled, H_2-only operation. In fact, the effect of adding hydrogen appears, in terms of engine efficiency, to be very nearly that of simply extending the trend of gasoline-engine efficiencies to lower equivalence ratios (e.g., $\phi \cong 0.3$) than would be possible with conventional gasoline operation ($\phi \cong 0.7$). Part-load (low ϕ) efficiencies are much higher (e.g. 50%) with such mixed-fuel operation than with conventional gasoline operation. This gain is typically cited for hydrogen engines. Large efficiency gains may not, however, maintain when hydrogen or mixed-fuel operation is compared with stratified-charge operation using gasoline only.

With H_2/gasoline as fuel, naturally-aspirated, reciprocating engines need not pay the maximum power-output (IMEP) penalty associated with H_2-only operation (see Section 2.2.1); at full load, the engine is a normal gasoline engine. Mean effective pressures vary with equivalence ratio from those of gasoline (at full load) to nearly those of hydrogen at low load (low equivalence ratio). At the lowest equivalence ratio tested by JPL (Ref. 10), the fuel is 50% H_2 by <u>weight</u>, i.e., about 97% by volume, and MEP is nearly that for H_2 at the same ϕ.

This allows quality-control rather than engine throttling. Below $\phi \cong 0.5$ (approximately 50% full load), NO_x emissions are reduced to the level equivalent to 1978 E.P.A. standards (0.4 gm/hp-hr) or below. However, unburned hydrocarbon emissions remain too high under all conditions (> 0.4 gm/hp-hr) and dictate a need for exhaust-gas conditioning (e.g., catalytic conversion). The problem with excessive unburned hydrocarbon emissions appears to be inherent in this type of engine; the only additional problems inherent in this type of engine are connected with the practicalities of generating hydrogen from gasoline (equipment development, cost, volume, weight, and reliability).

Further consideration of the hydrogen generator developed by JPL and its significance as an add-on to the reciprocating engine was not consistent with the eventual de-emphasis of reciprocating engines in the present program.

1.6 - HYDROGEN/OXYGEN ENGINES

Reciprocating H_2/O_2 engines have been operated during development of small auxiliary power systems for NASA aerospace applications in the 1960's (Ref's. 6,7) and more recently in the U.S. and Japan with the intention of reducing pollutant emissions (Ref's. 5, 8). Some results for NASA aerospace power units (Ref. 6) have been discussed in an earlier report (Ref. 13), and the hardware and data of Ref. 7 are very similar. Detailed data-taking was not included in the efforts reported by Dieges and Underwood (Ref. 5) and those of Furuhama (Ref. 5) are insufficient at present for meaningful fundamental interpretation since only brake output powers, efficiencies, etc. are reported and the reported data are not well-defined.

Flame temperatures in stoichiometric H_2/O_2 are too high (e.g., 6000 F) to be attractive for reciprocating-engine operation, and H_2/O_2 mixtures are diluted from stoichiometric proportions. Excess O_2, excess hydrogen, recycled H_2O (product), or other gases are possible diluents. Of these, only excess H_2 has apparently been used (Ref's. 5-9). In this context, however, H_2/air engines can be considered as H_2/O_2 engines with N_2 as a diluent.

Noble-gases are attractive diluents since the high specific-heat-ratios of monatomic gases contribute to improved theoretical cycle efficiencies. However, cycle calculations for argon and helium as diluents have suggested problems with autoignition if these are added to H_2/O_2 as diluents (Ref. 28). This is because the high ratio of specific heats of these is necessarily accompanied by low molar specific heats (C_p, C_v). Therefore, higher temperature rises occur during the compression stroke, promoting autoignition. While this effect might conceivably be turned to advantage in an attempt at Diesel-like H_2/O_2 injection operation, no such attempts are known to have been made. It is noteworthy that H_2/O_2 engines (Ref's. 5, 6, 7) have experienced ignition difficulties, and it is possible that noble-gas diluents would prove useful after further development.

H_2O vapor is an attractive diluent except for its low ratio of specific heats (which degrades cycle performance) and potential problems with condensation on relatively cool engine-system parts. As a diluent, H_2O is, however, especially attractive in situations where purposeful, complete exhaust-gas condensation is desirable (e.g., underwater).

Excess O_2 or excess H_2 as diluents are comparable in thermodynamic effect on cycle performance, both being diatomic gases. In addition, to reduce mixture temperature to practical levels by dilution similar numbers of diluent moles are required whether O_2 or H_2 are in excess, since the molar heat capacities of the two gases are very similar. H_2 enjoys only a slight edge owing to a somewhat higher effective heat capacity (including dissociation effects). The major argument against use of excess O_2 as a diluent is apparently the hazard encountered (Ref. 28). Hot (wet) oxygen as a combustion product could lead to combustion of lube oil through leakage to the crankcase. Other corrosion and oxidation problems can be anticipated as well. Therefore, excess H_2 is apparently the preferred H_2/O_2 diluent.

Since dilution is required to render an H_2/O_2 reciprocating engine practical, it is to be expected that cycle efficiencies and power output should not be greatly different than with H_2/air reciprocating engines. A detailed comparison of H_2/O_2 and H_2/air engine has not been found in the literature nor carried out as part of the presently-reported effort. Isolated calculations for H_2/O_2 engines have been made, however, and the theoretical performance indicated is comparable to or below that theoretically calculated for H_2/air engines (Ref's. 5, 28). The few experimental data available for H_2/O_2 reciprocating engines, likewise, indicate lower performance, (efficiency, MEP) than would be expected for comparable H_2/air engines.

It is concluded that, with the exception of the potential for eliminating NO_x emissions and the possibility of condensing products of combustion rather easily, H_2/O_2 reciprocating engines of conventional type offer little, if any, advantage as power plants. With the two exceptions cited, H_2/air engines have shown superior performance theoretically and experimentally.

Only highly specialized applications are likely candidates for H_2/O_2 reciprocating engines, notably those where combustion air is not readily available. The modest effort devoted so far to H_2/O_2 engines indicates no fundamental problems with such engines. However, work to date has indicated a substantial need for further development of fuel handling and injection systems and ignition systems as well as leaving uncertain what problems might arise regarding long-term materials compatibility and reliability.

1.7 - SUMMARY

There is no evidence of successful compression-ignition ("Diesel") engines operated on pure hydrogen, and significant experimental results dictate against this mode of hydrogen-fuel use in reciprocating engines. Just as in so-called "gas-engine" modifications of Diesel engines, however, hydrogen-gas can be used in compression-ignition engine hardware if ignition is provided by small supplementary "pilot" charges of hydrocarbon fuel, by a glow plug, etc.

Spark-ignition engines (in contrast to compression-ignition engines) have been demonstrated in numerous cases to operate with pure hydrogen fuel at substantially improved efficiencies compared with conventional, spark-ignition gasoline engines*. However, problems of power loss, rough-running, flash-back, and excessive NO_x emissions are encountered with hydrogen fuel under various operating conditions. The latter three problems can be overcome by water injection and/or exhaust-gas recirculation and/or by burning only lean mixtures (e.g., 60% of stoichiometric or less). These "fixes", however, further decrease power output compared, say, with that from a similar gasoline-fueled engine without "fixes". Therefore, only at substantial loss in maximum power and specific power (e.g., 50%), can hydrogen be used

* Efficiency improvements arise from the use of mixture-ratio ("quality") control with hydrogen rather than throttling ("quantity"-control) as with gasoline. A definitive comparison between quality-controlled hydrogen and quality-controlled gasoline (via "stratified charge") has not been made in this study or, apparently, elsewhere.

at improved efficiency and low NO_x by fairly simple modification of naturally-aspirated engines.

All of the cited problems can be overcome by substantial engine modification to allow for direct-cylinder-injection except possibly for problems with (1) rough-running due to high rates of pressure rise during combustion and (2) excessive NO_x emissions. Direct cylinder injection can be at high-pressure (near the end of the compression stroke) or at low pressure (early in the compression stroke). The requisite engine-coupled compressors for hydrogen compression have not been demonstrated and are likely to prove a troublesome problem. Similarly, pumps and vaporizers to allow the use of LH_2 have not been demonstrated and will probably present problems with respect to matching demand, stability, etc.

Injection at high pressure (concurrent with combustion or nearly so, as in Diesel engines) can probably eliminate rough-running caused by high rates of cylinder pressure rise. This probability has not been definitively demonstrated, however, except in experiments for which efficiency and/or power output was abnormally low (for currently debated reasons). High-pressure injection has also been demonstrated to allow low NO_x emissions but, again, concurrent with lower efficiencies and power outputs than expected. These effects are not rationalized at present, and further research and development concerned with injection and combustion systems is indicated. The practical potential for minimizing NO_x formation by injector-system design has never been evaluated and such evaluation represents a substantial R & D task. The present lack of development of high-pressure injection severely constrains an evaluation of the practical potential and problems of such injection.

In contrast, low-pressure hydrogen injection (early in the compression stroke) has demonstrated increases in power output compared with naturally-aspirated hydrogen engines. This improvement over naturally-aspirated engines is unfortunately, at the expense of combustion "knock" and, though unmeasured, probably also at the expense of high NO_x emissions. Both of these problems tend to worsen as near-stoichiometric, full-power operating conditions are approached. These two problems can be abated at part-load by the lean-burning achievable readily with hydrogen. However, as with high-pressure injection, there are no data available regarding the potential for decreasing knock and NO_x emissions by injector-system design (e.g., to provide stratified charge). This fact severely compromises evaluation of the low-pressure alternative for direct cylinder injection.

The likelihood of abating NO_x emissions and high rates-of-pressure-rise (combustion knock) by water injection and/or exhaust-gas recirculation during high-power-output operation has not been investigated in either high-pressure or low-pressure cylinder-injection engines. These means are probably effective though accompanied by moderate loss in output power.

Spark-ignition engines have been successfully operated on mixtures of hydrogen and hydrocarbon fuels (liquid and gaseous). Relative to conventional use of pure hydrocarbon fuel, the sole advantages of this mode of operation are improved engine efficiency at part load (much as for pure hydrogen operation) and substantially reduced NO_x emissions at part load. Excessive emission of unburned hydrocarbons is an inherent problem, presumably solvable by catalytic conversion or other exhaust-gas conditioning. Development of a practical hydrogen generator is a major problem.

Hydrogen/oxygen reciprocating engines have been little investigated and, therefore, their problem areas are poorly defined. However, there is no basis for expecting significant performance gains from this type of engine compared with hydrogen/air engines except in specialized applications for which the availability of air, the elimination of NO_x, and/or the condensibility of exhaust products are prime factors.

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APPENDIX 1.A - APPROXIMATE INFLUENCE OF CYLINDER FUEL INJECTION ON OTTO-CYCLE EFFICIENCY

Approximate analysis shows that, while cylinder fuel injection (CFI) of gaseous fuel can be expected to increase Otto-cycle efficiency, the effect is small for reasonable fuel-injection conditions. This fact is made clear by analyzing the effect of fuel injection on the approximate "air-standard" Otto-cycle. This approximation, while known to be unrealistic for overall cycle-performance calculations, is used here only in order to account for what is expected to be the major impact of fuel injection on the theoretical cycle, i.e., a decreased number of reactant moles compressed in the engine cylinder during part or all of the compression stroke. To shorten manipulations, only differences between simple cycles are analyzed.

Figure 1.A-1 is a schematic P-V diagram showing:

(i) a naturally-aspirated (Otto) air-standard cycle (1-2-3-4)

(ii) a supercharged (CFI) air-standard cycle (1-x-x'-2'-3'-4') with 1-n moles of gaseous fuel injected into 1 mole of air at volume V_x, raising the cylinder pressure between x and x'.

For comparison of the efficiencies of the two cycles, it is noteworthy that air-standard-Otto-cycle efficiencies depend only on the compression ratio, r:

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \qquad \text{(A-1)}$$

With this relation as a basis, it is clear that:

$$\eta_{1-2-3-4} = \eta_{Otto} = \eta_{1'-2'-3'-4'} = \eta_{EP}$$

where EP symbolizes an "elevated-pressure" Otto cycle. Assuming that each cycle produces net work, W, from the addition of the same external heat, Q_{in}, during process (2-3) or (2'-3'), then:

$$\eta_{Otto} = \eta_{EP} = \frac{W_{net,EP}}{Q_{in}} \qquad \text{(A-2)}$$

Considering the CFI cycle (1-x-x'-2'-3'-4'):

$$\eta_{CFI} = \frac{W_{net,CFI}}{Qin} = \frac{W_{net,EP}}{Qin} + \frac{\Delta W_{net}}{Qin}$$

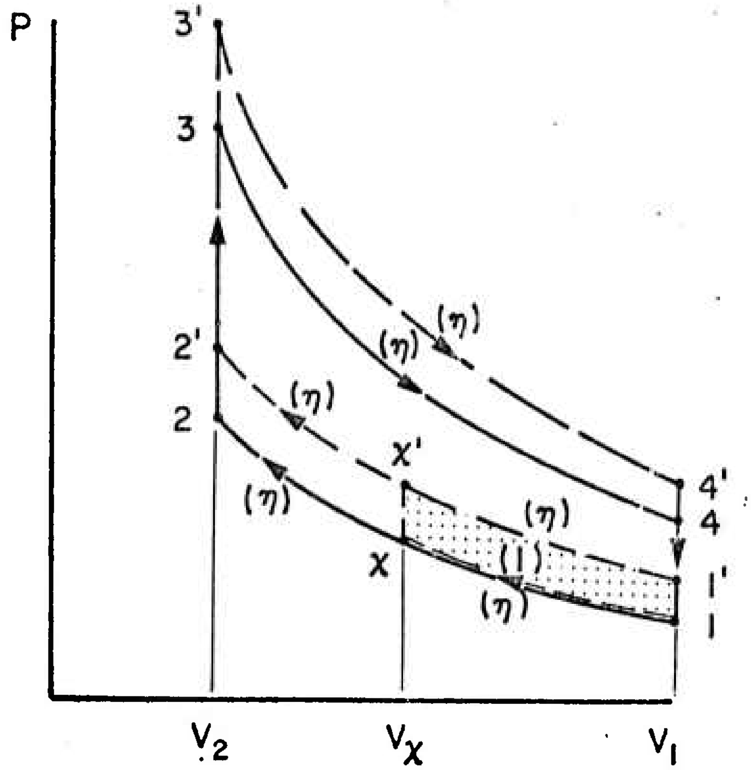

FIGURE 1.A-1: Schematic P-V Diagram for Cylinder Fuel Injection

$$= \eta_{Otto} + \frac{\Delta W_{net}}{Qin} \qquad (A-3)$$

The changes in net work, ΔW_{net}, between the CFI and the EP cycle results from decreased compression work for the CFI cycle:

$$\Delta W_{net} = (n) w_{1'-x'} - (1) w_{1-x}$$

where w = compression work input/mole

For isentropic compression of a perfect gas, then:

$$\Delta W_{net} = nC_v T_1'(r_x^{\gamma-1}-1) - C_v T_1 (r_x^{\gamma-1}-1)$$

$$= (nC_v T_1' - C_v T_1)(r_x^{\gamma-1}-1) \qquad (A-4)$$

where $r_x \equiv \frac{V_1}{V_x}$

Assuming that injected fuel is supplied from a large reservoir of perfect gas at fixed temperature (T_F), pressure (P_F) and molar volume (V_F), the First Law applied to an adiabatic injection process yields:

$$(n) C_v T_{x'} = (1) C_v T_x + (n-1) C_v T_F + (n-1) P_F V_F$$

$$= C_v T_x + (n-1) C_p T_F$$

or, for isentropic compression:

$$nC_v T_1'(r_x^{\gamma-1}) = C_v T_1 (r_x^{\gamma-1}) + (n-1) C_p T_F \qquad (A-5)$$

Multiplying (A-5) by $(r_x^{\gamma-1}-1)$ and rearranging:

$$(nC_v T_1' - C_v T_1)(r_x^{\gamma-1} -1) = (n-1) C_p T_F \frac{r_x^{\gamma-1} -1}{r_x^{\gamma-1}} \qquad (A-6)$$

Substituting (A-6) into (A-4):

$$\Delta W_{net} = (n-1) C_p T_F (1 - \frac{1}{r_x^{\gamma-1}})$$

Therefore, from (A-3) with $Q_{in} \equiv (n-1)\psi$ and $\Psi \equiv$ chemical energy release per mole:

$$\eta_{CFI} = \eta_{Otto} + \frac{C_p T_F}{\Psi}(1-\frac{1}{r^{\gamma-1}}) \qquad \text{(A-7)}$$

The second term represents the increased efficiency (net work) deriving from gratuitous injection-gas compression. This contribution can be viewed as deriving from work output available (without corresponding work input) by expanding (from V_x to V_1) the (n-1) moles of fuel gas injected with an initial enthalpy of C_pT_F.

From Eq. (A-7) one sees that, due to the injection of gaseous fuel, the CFI-cycle efficiency should be greater than that of the naturally aspirated Otto-cycle whenever $r_x = \frac{V_1}{V_x} > 1$, i.e., for any injection timing beyond bottom-dead-center. The maximum increase in efficiency occurs when r_x = maximum $\frac{V_1}{V_2}$ r, i.e., when injection is at top-dead-center:

$$\eta_{CFI,max} = \eta_{Otto} \, (1 + \frac{C_p T_F}{\Psi}) \qquad \text{(A-8)}$$

since $\eta_{Otto} = 1- \frac{1}{r^{\gamma-1}}$

Numerically, (A-8) leads to the conclusion that efficiency increments due to CFI are, at most, small. Table A-1 shows some extreme effects of injection on efficiency and indicates that injection from compressed gas bottles (in laboratory testing) might be expected to increase efficiencies by about 4%, i.e., 1 or 2 percentage points in efficiency.

TABLE A-1: H_2 FUEL INJECTION EFFECTS ON AIR-STANDARD OTTO-CYCLE EFFICIENCIES.

T_F	$\frac{C_p T_f}{\Psi}$	η_{Otto} (Assumed)	$\eta_{CFI,max}$	$\Delta\eta_{CFI}$
80F	.037	.3	.31	.01
		.5	.52	.02
540F	.068	.3	.32	.02
		.5	.53	.03
1000F	.099	.3	.33	.03

In practice, the slight efficiency gains indicated in Table A-1 would be lost if a mechanical compressor were required for raising the gas to suitably high injection pressures. Table A-2 shows, as fractions of the lower heating value (LHV) of H_2, the mechanical compressor work, W_c, required by various compressor inlet temperatures (T_i), pressures (P_i), adiabatic compressor efficiencies (η_c), and discharge (injection) pressures (P_o).

TABLE A-2: ENGINE EFFICIENCY DECREMENTS DUE TO MECHANICAL COMPRESSION OF H_2

η_c (-)	T_i (F)	P_i (Atm)	P_o (Atm)	W_c/LHV $=\Delta\eta_c$ (-)
0.60	-280	1	10	-.019
		1	30	-.033
		1	100	-.055
		10	100	-.019
0.60	80	1	10	-.056
		1	30	-.099
		1	100	-.16
		10	100	-.056
0.80	-280	1	100	-.041
0.80	80	1	100	-.12

The values for W_c/LHV shown in Table A-2 can be interpreted as approximate values of the decreases in engine-system efficiency ($\Delta\eta_c$) which would result from using a portion of the engine power output to drive a fuel-gas compressor. These values are large enough to more than offset the small efficiency gains of CFI operation (Table A-1). The efficiency decrements are also substantial compared with brake efficiencies anticipatable with H_2 engines (e.g., η_B=0.20 to 0.40). Compression-work values are large enough to represent a substantial potential for innovative H_2 compression schemes.

SECTION 2

HYDROGEN-FUELED GAS TURBINES

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TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

SECTION 2 - HYDROGEN-FUELED GAS TURBINES

2.1 - INTRODUCTION

The application and technology of the conventional open-Brayton-cycle gas turbine is well developed for both liquid and gaseous petroleum fuels. A major impetus has come from the gas-turbine's unique applicability to jet propulsion. Power-plant research, development, design and fabrication methods for aircraft gas turbines have spun-off into the substantial use of aircraft-derivative power plants for stationary applications such as pipe-line and other industrial-compressor drives, peak electrical power generation, and marine propulsion. Many of these applications would not be feasible at present except for much of the development cost of gas turbines having been justified by military-aircraft needs.

The wide application of gas turbines derives from several advantages of such power plants relative to the more traditional types such as Diesel and gasoline engines or steam turbines:

- low weight and volume per unit of output power
- rapid start and development of full power
- minimum accessories and support services required
- multi-fuel capability
- ease of replacement and major overhaul
- reasonable thermal efficiency

Traditional gas turbine problems include:

- lower efficiencies than alternative power plants, especially at part-load
- high capital cost
- limited hot-section lifetimes due to steady, high temperatures at peak power; thermal stresses during cycling; corrosion from fuel and air contaminants
- large air flows required (ducting, air filtering, etc. in stationary or marine applications)
- salt-water ingestion in marine applications

Two major considerations bear on the potential problems and practicality of hydrogen as a gas-turbine fuel. First, any unique values of hydrogen-fueled turbines must be considered; as in the past, these will tend to dictate directions for fruitful early application and will also provide impetus for spin off into still other applications. Second, hydrogen usage may be justified on some other basis than its unique value as a gas-turbine fuel. In this case, the current widespread use of gas turbines mandates an evaluation of the probable impact of hydrogen fuel on conventional gas-turbine practice. To deal with both these considerations, one must first overview the distinctive features of hydrogen in the context of gas-turbine use.

2.2 - DISTINCTIVE FEATURES OF HYDROGEN AS A GAS-TURBINE FUEL

Deriving from the chemical, thermodynamic and combustion properties of hydrogen/air mixture, the following features distinguish hydrogen as a gas-turbine fuel:

- high mass energy density (except as hydride)
- low volumetric energy density
- low molecular weight; gaseous state at normal temperature and pressure
- broad flammability limits
- high reaction rates, volumetric heat-release rates
- low radiative emission from flame and combustion products
- condensibility of benign major product of combustion (H_2O)
- purity (depending on source, e.g., electrolytic)
- high heat-sink capacity (especially as cryogenic LH_2)

The first three items bear most strongly on storage and handling systems; the remainder on the gas-turbine itself.

The combustion-related features of hydrogen (flammability limits, reaction rates) contributed to earlier extensive investigation of hydrogen as a fuel (Ref's. 1,2). Hydrogen allows substantial extension of altitude limitations on turbojet aircraft (Ref's. 3-6), and, therefore, allows new aircraft missions as well as the possibility of combustion in supersonic air streams for ramjets, etc. Extensive bibliographies of earlier investigations appear in References 1 and 7.

The high mass energy density has earlier (Ref. 3) and also recently (Ref's 8,9) been seen to allow more favorable range/payload options than conventional hydrocarbon fuels in aircraft for which fuel weight is a large fraction of take-off weight (such as cargo transports).

Cryogenic hydrogen's heat-sink capacity is potentially useful as a source of auxiliary power _via_ ambient or exhaust-gas heat recovery, as an engine coolant, and for cooling or condensing exhaust products. Other peripheral low-temperature uses are possible such as air separation, superconductor cooling (heat-shielding), electronic (e.g., microwave) cooling, etc. Except for superconductive applications and air liquefaction, cryogenic hydrogen's low normal boiling point ($-420^{\circ}R$) does not appear, however, to offer qualitatively unique returns compared with other cryogenic fuels, notably liquified natural gas (normal boiling point: $-260^{\circ}F$).

The condensibility and benignity of the major combustion product, water, have been advanced as unique values for hydrogen as a fuel in all pollution-sensitive situations and especially in confined quarters, e.g., underwater, interior mobile transport, etc. where release of combustion products is undesirable.

The purity of hydrogen produced, for example, by electrolysis is not unique, in that other fuels can be purified, though at substantial additional cost. Of major interest is the likelihood of fuel hydrogen's being devoid of sulfur, light metals and their salts, and various trace metals all of which contribute to hot-section corrosion in conventional hydrocarbon-fueled gas turbines.

Despite this array of distinctive positive features, there is no apparent, qualitatively new application of hydrogen-fueled gas turbines, with two exceptions:

- a gas turbine powered indirectly by only electricity and water (e.g., H_2/O_2 by electrolysis with H_2O dilution)

- a zero gas-emissions turbine (H_2/O_2 with condensed products)

Both of these distinctive possibilities are feasible technically and have been proposed earlier (Ref's. 10,11,12), but have not apparently been demonstrated.*

Aside from H_2/O_2 gas turbines, hydrogen-fueled gas turbines offer gains which are quantitative rather than qualitative relative to conventional hydrocarbon/air gas turbines.

It is possible that these gains are sufficient to discount current and future economic and practical disadvantages of hydrogen as a fuel. Hydrogen as a gas-turbine fuel warrants investigation for this reason and also by virtue of a possible commitment to hydrogen fuel for reasons extraneous to gas turbines per se.

The remainder of this technical report is devoted to uncovering the demonstrated and implied problems of hydrogen as a fuel in conventional, hydrogen/air gas turbines. Since the conventional open-cycle gas turbine is inherently a multi-fuel power plant, one might expect that any major impact would derive from practical rather than fundamental theoretical considerations. However, there is an apparent lack of general-use theoretical treatment of hydrogen gas turbines in the readily-available literature. Therefore, consideration is given in the following section to such a treatment, in order to provide perspective. More practical concerns and technical problems which derive from them are confronted later in the report.

2.3 - IMPACT OF HYDROGEN FUEL ON GAS-TURBINE PERFORMANCE

2.3.1 Overall Engine Cycle

2.3.1.1 Simple, Ideal Cycle (Gaseous Fuel)

To estimate the extent of hydrogen's influence on the basic open-gas-turbine cycle, several thermodynamic cycle calculations were carried out. The ideal, simple open Brayton cycle, was considered.

*An uncompleted program for an H_2/O_2 turbine for electric power generation on a large scale (11 MW_e) was initiated by North American Rockwell (Ref. 13).

Table 2-1 specifies the conditions for these calculations.

TABLE 2-1: SPECIFICATIONS FOR HYDROGEN/AIR BRAYTON-CYCLE CONSIDERATIONS
- Perfect gases and homogeneous perfect-gas mixtures - Thermodynamics properties from Ref. 14 - Species: N_2, O_2, H_2, H_2O (complete combustion, no dissociation) - Inlet conditions: 1 atm - Exhaust conditions: 1 atm - Lower heating value of H_2: 51,600 BTU/lb - All components are loss-free

Thermal efficiency, specific air rate, specific work, and exhaust temperatures were calculated at two overall pressures ratios (OPR) of 6.0 and 20.0 roughly representative of industrial and aircraft gas turbines. A range of equivalence ratios ($0.1 \leq \Phi \leq 0.5$) was also considered.

Table 2-2 shows some of the results, along with comparable hydrocarbon fuel results calculated on the same basis. The results illustrate the fact that substitution of H_2-gas for hydrocarbon fuel with fixed turbine inlet temperature (TIT) or temperature change across the combustor leads theoretically to a modest decrease in cycle efficiency (e.g., 2 to 3%) and an increase in specific work (e.g., 3 to 5%).

These changes arise from changes in the thermodynamic properties of the combustion products, notably a lower average molecular-weight with hydrogen (by approximately 6% at TIT = $3000^{\circ}R$, OPR = 20) and a larger number of product moles per mole of air handled. The effects of different molar specific heats and ratios of specific heats are not great for the two different fuels. Thus, for a given turbine inlet temperature and overall pressure ratio, the increased theoretical specific work with hydrogen fuel derives largely from the increased number of moles of product per unit

TABLE 2-2: ESTIMATED INCREMENTS IN EFFICIENCY AND SPECIFIC POWER WITH HYDROGEN SUBSTITUTION FOR LIQUID HYDROCARBON

Max. Efficiency:	0 to -3%
Specific Work (HP/LB AIR):	+5%

For simple, ideal cycle, assuming no fuel-gas compression-work loss mass of air handled. However, the larger number of moles and nearly equal molar specific heat require that more fuel heating value must be supplied with hydrogen to achieve the specified turbine inlet temperature. This factor more than discounts the increased specific work with hydrogen and results in a small theoretical efficiency decrease relative to typical hydrocarbon fuels.

These rough estimates of the impact of hydrogen use on gas-turbine performance are in general accord with prior analysis of turbojet performance with hydrogen fuel (Ref. 15). English (Ref. 16) has documented a loss of efficiency (ca. 2-3%) and gain in specific thrust (ca. 3-5%) with hydrogen over a range of turbojet operating conditions. More recently, similar conclusions were reached from analysis of turbofan engines (Ref. 17). Thus, within about 5% accuracy, it appears reasonable, theoretically to expect about the same basic thermodynamic performance using gaseous hydrogen as with more conventional hydrocarbon fuels. Figure 2-1 shows a summary of recent gas-turbine efficiency (SFC) data as a function of output power with marine-steam-turbine and diesel-engine data included for reference.

2.3.1.2 Elevated fuel temperature

Unlike conventional liquid gas-turbine fuels, hydrogen can be admitted to a gas-turbine combustor over a wide range of temperatures. In one extreme, brief experiments have demonstrated that injection of cryogenic liquid hydrogen (ca. 40°R) is possible. In the other extreme, gaseous hydrogen can be heated to elevated injection temperatures without degradation (e.g., 1000°F); fuel pyrolysis, coking, etc. are not problems with hydrogen.

In thermodynamic terms, the effect of elevated fuel temperature is to require less fuel to be burned if a fixed combustor outlet temperature or turbine inlet temperature (TIT) is not to be exceeded. For specified TIT and other turbine conditions and for 1000°F hydrogen supply, the hydrogen flow rate may be decreased in the ratio:

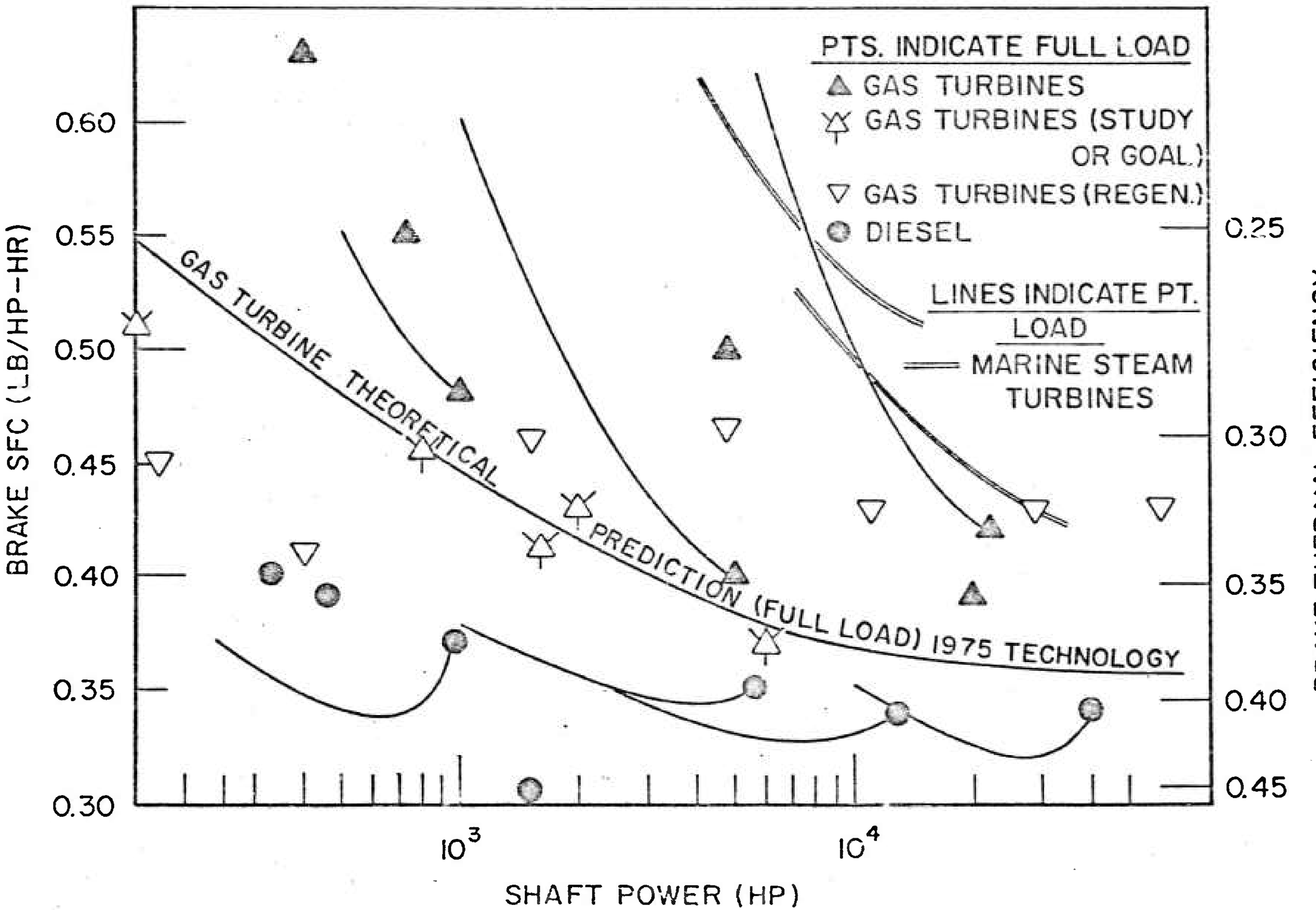

FIGURE 2-1: Summary of Recent Gas-Turbine Efficiency/ Power Output Data

$$\frac{LHV - [(h_{H_2})_{1000^{\circ}F} - (h_{H_2})_{80^{\circ}F}]}{LHV} \cong \frac{48,400}{51,600} = .94$$

i.e., by about 6%. The cycle efficiency may, therefore, be increased by about 6%. Thus, if heated GH_2 is used as fuel, the efficiency loss of hydrogen relative to hydrocarbon fuel (see preceding section), can be more than recovered. Engine cooling or exhaust-gas heat recovery are possible means for heating the hydrogen fuel, though not without practical problems.

Conversely, GH_2 (or LH_2) temperatures below ambient represent potential losses of engine efficiency (see Section 2.4.5.5, below).

2.3.1.3 Improved temperature distribution

Maximum cycle temperature is usually considered equivalent to average turbine inlet temperature (TIT). Turbine performance is limited by turbine-materials technology. One of the consistent trends in turbine development has been toward higher cycle temperatures (Ref's. 18-20). These trends have been made possible by improved material technology and hot-section cooling. The trend of approximately 40 to 60°F/yr in TIT for aircraft gas turbines corresponds, for example, with a trend of 10 to 30°F/yr increase in creep-strength temperatures for blade and vane alloys (Ref. 21). Beyond the apparent cycle-temperature limits imposed by turbine-material technology, other practical effects further limit turbine inlet temperatures. Non-uniform temperature distribution of the gases entering the turbine from the combustor requires turbine operation at average temperatures which are lower than the maximum otherwise sustainable by the turbine. Minimizing temperature non-uniformities, therefore, can allow higher average turbine inlet or cycle temperatures without increases in maximum local inlet temperature.

Greater temperature uniformity also contributes to less thermal distortion of turbine shrouds. This is especially true of the first stage, thus permitting smaller blade-tip clearances which result in a turbine efficiency increase. Small turbines are more influenced by this effect than large ones (Ref. 22).

Temperature non-uniformity at turbine inlet can be characterized quantitatively by the "temperature variation ratio" (TVR) or "pattern factor":

$$TVR = \frac{(T_4)max - T_3}{(T_4)avg - T_3}$$

where T_4 and T_3 are, respectively, turbine and combustor inlet temperatures.

Few data are apparently available regarding TVR's with hydrogen fuel. Some early experiences in combustor testing indicated little if any improvement with hydrogen compared with liquid hydrocarbon (Ref. 23). However, some more recent data (Ref. 24) show a TVR value (ca., 1.1) appreciably lower with hydrogen than is common in production engines using hydrocarbon fuels (TVR = 1.2 to 1.4).

Such a difference in TVR implies a difference of 100-300^{o}F in peak temperature. Therefore, the possibility exists of a 100 to 300^{o}F increase in average TIT and a corresponding increase in performance without exceeding the previous peak temperature. At current military turbojet TIT's and OPR's a TVR decrease of 0.1 to 0.3 implies approximately 1 to 4% increase in efficiency and 5 to 15% in specific output (thrust or shaft power).* Improvements are less for higher-performance (high TIT, high OPR, high turbine and compression efficiencies) than for lower-performance units. Somewhat greater gains may be possible with further development; there is little evidence of hydrogen combustion development to data with emphasis on empirically improving TVR.

There is no experimental basis at present for judging to what extent decreased TVR with hydrogen could ease thermal-stress problems with turbine hot-section parts, thereby allowing even higher average turbine inlet temperatures. Evaluation of such aspects requires design studies beyond the scope of the present effort.

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*Cited improvements in efficiency or specific power are not concurrent, i.e., estimates are based on cycles optimized for either maximum efficiency or maximum specific power, respectively.

2.3.1.4 Decreased Corrosion

Hot-section corrosion, erosion and deposits also force some industrial gas-turbines to operate at lower turbine inlet temperatures than otherwise would be practical. As with TVR improvements, alleviation of these problems can allow higher operating temperatures and increased performance.

However, current gas turbines are usually operated with low sulfur fuels to minimize hot-section corrosion problems, sulfidation being a major concern. Substitution of low-sulfur fuel in industrial gas turbines has in some cases allowed on the order of 100°F increases in turbine exhaust temperature (Ref. 25), corresponding to 150° to 250°F increase in TIT. Thus, compared with high-sulfur fuels, hydrogen might reasonably allow 150 to 300°F increases in TIT depending on OPR; concurrent efficiency and specific power increase might, thereby, approach those cited in the previous section, i.e., 1-4% in efficiency, 5-15% in specific power.

In marine applications, even with hydrogen fuel, ingested salt water will probably contribute to hot-section corrosion.

2.3.1.5 Reliability

Gas turbine reliability and performance are interrelated as a consequence of each being strongly dependent on TIT. Sensitive components are the combustor parts especially the combustor liners as well as the first several turbine stages.

Previously cited performance gains possible with hydrogen (based on TIT increases) can be traded-off with reliability. In high-performance aircraft engines, an order-of-magnitude increase in operating time (at maximum thrust) might be expected from a 50 to 100°F decrease in maximum TIT (Ref. 26). Such an estimate can also be made based on stress-rupture data for typical hot-section alloys (see Fig. 2-2). On this basis, the improved performance possible with hydrogen (due to lower TVR and, possibly, decreased hot-section corrosion) could potentially be traded-off for substantially increased reliability. Operating times might be increased by a factor of ten to a hundred.

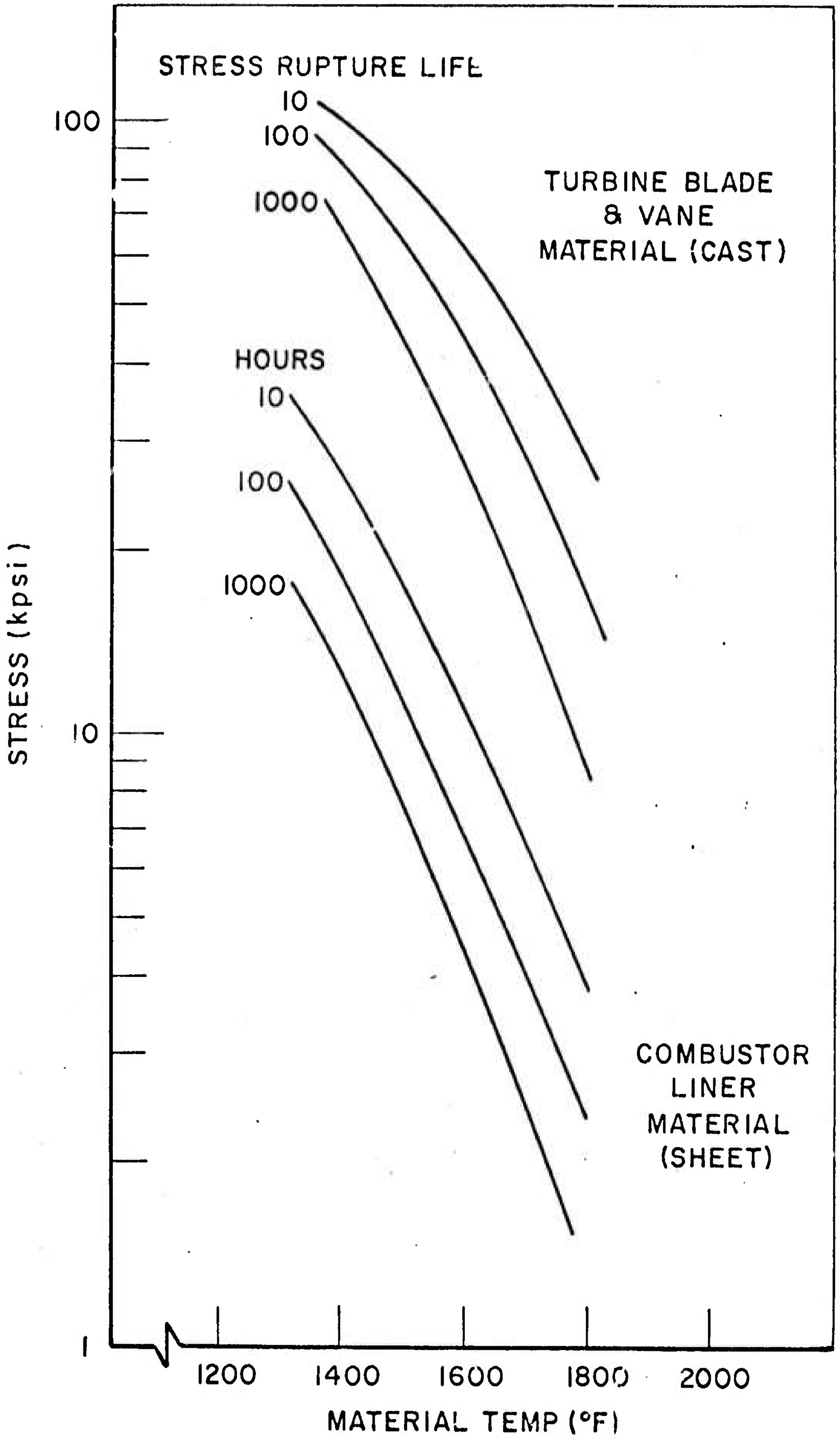

FIGURE 2-2: Hot-Section Stress/Rupture Data

2.3.1.6 Pollutant emissions

Elimination of fuel carbon by the use of hydrogen eliminates CO, CO_2 and unburned hydrocarbons as pollutants (except for lube-oil pyrolysis products), NO_x and H_2O remaining as major concerns. Abatement of NO_x emissions with current conventional fuels is a relatively new concern in gas-turbine development and is the subject of active research and development; (e.g., Ref. 27); thus, a reasonable baseline emissions level characterizing NO_x emissions with hydrocarbon fuels is not available. H_2O is likely as a significant "pollutant" only at high-altitudes where it may disrupt normal atmospheric chemistry (Ref's. 28,29).

Recently some comparative tests and studies of hydrogen vis-a-vis hydrocarbon fuels have been made with the aim of comparing NO_x emissions with the two fuels (Ref's. 30,31). Other recent work involves the addition of hydrogen to promote stability and to decrease emissions from hydrocarbon combustors (Ref. 32). However, as mentioned previously with regard to TVR, programs have not apparently been directed specifically at developing minimum-NO_x combustors which take maximum advantage from the unique combustion properties of hydrogen (Ref. 27).

Thermodynamically, little difference can be expected in NO_x formation between hydrogen and hydrocarbons when both fuels are burned to the same temperature. At a given, practical fuel/air mixture ratio or equivalence ratio, hydrogen theoretically yeilds a somewhat higher flame temperature (Fig. 2-3) and, therefore, somewhat more NO_x than JP-4 fuel; the curves labelled "equilibrium" on Figure 2-4 show this effect. However, hydrogen allows operation at equivalence ratios approximately 12% lower if given combustor and turbine inlet temperatures are specified. This can discount the comparison shown in Fig. 2-4 and yield approximately the same equilibrium NO_x with hydrogen as with JP-4 at a given combustion temperature.

Peak, equilibrium NO_x concentrations as calculated are not, however, particularly useful characterizations of real NO_x emissions. It is very well established that combustor temperatures

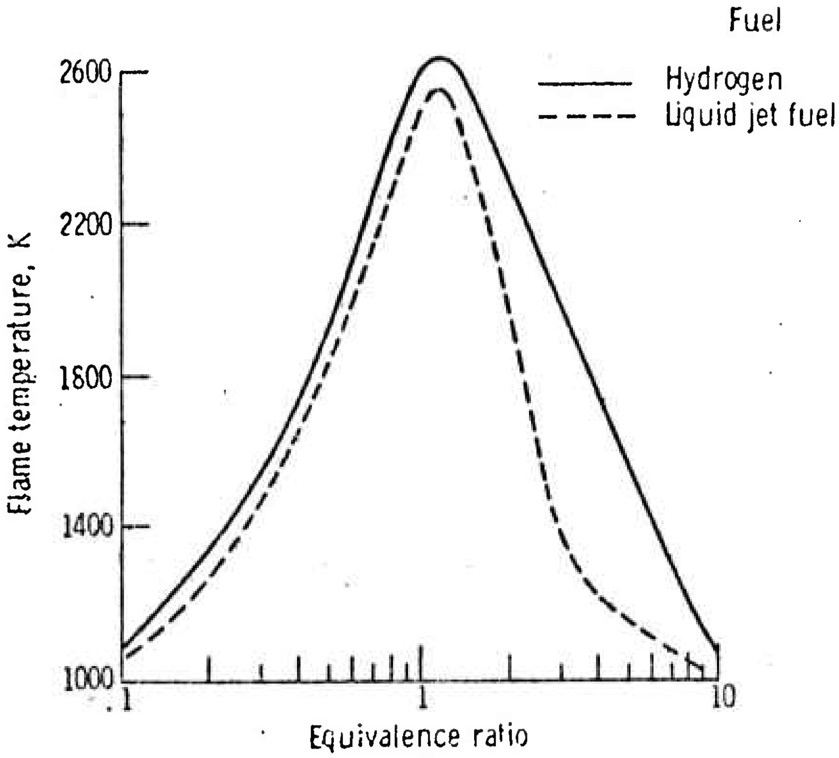

FIGURE 2-3: Theoretical Flame Temperatures (Ref. 30)

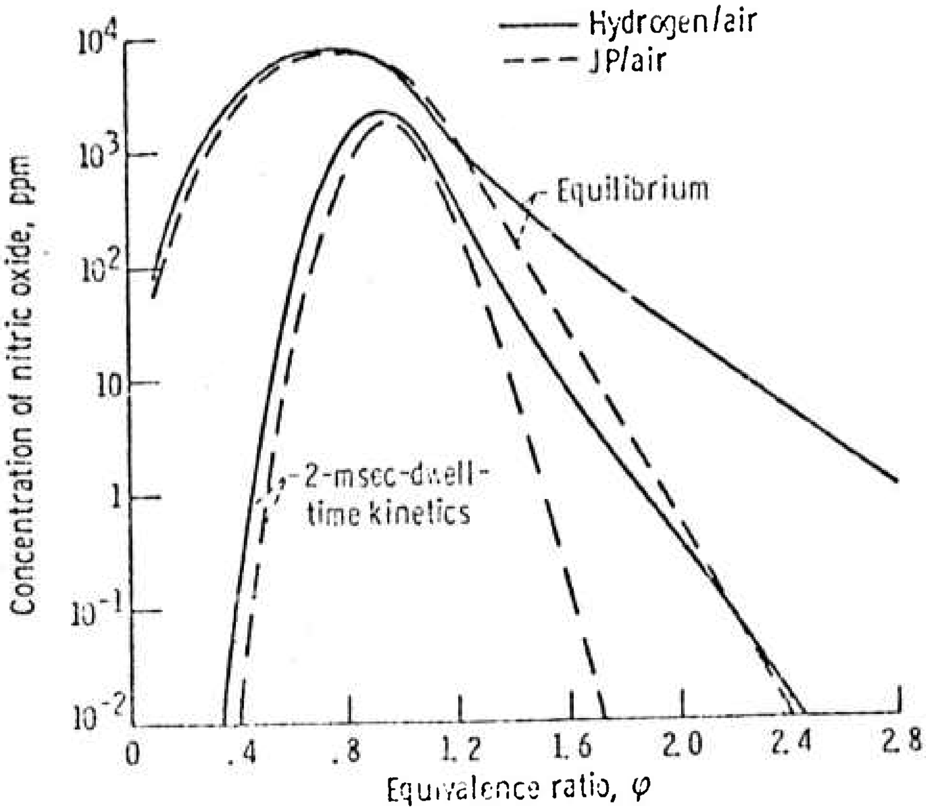

FIGURE 2-4: Theoretical Nitric-Oxide Concentrations (Ref. 27)

can be too low to allow finite chemical-reaction rates to yield near-equilibrium NO_x concentrations, especially with lean primary combustion zones. This is true with both hydrogen and hydrocarbon fuels. The reaction-rate aspects of NO_x formation with hydrogen are dealt with farther below in Section 2.3.2.2.

Emissions of H_2O into the troposphere and/or stratosphere in large quantity can impact on atmospheric chemistry and lead to classification of H_2O as a pollutant. While not a pollutant in the sense of chemical toxicity, such H_2O can interplay with upper-atmospheric ozone to alter atmospheric absorption of solar and terrestrial radiation and therby alter mean temperature on the earth. Models of such interplay were made in studies of the environmental impact of large-scale, commercial supersonic flight at high altitude (Ref. 28). Detailed atmospheric models are still of uncertain validity (e.g., current controversy over impact of halogenated hydrocarbon pollutants in the upper atmosphere). It has been suggested that tropospheric effects of H_2O emissions would be small since it is assumed that H_2 aircraft would operate predominately in the stratosphere. However, stratospheric effects are uncertain and better definition of stratospheric dynamics and chemistry is required (Ref. 28). Nonetheless, it appears that large-scale aircraft emissions of H_2O may become significant if high-altitude aircraft use continues to grow (e.g., to several thousand aircraft. Large-scale use of hydrogen fuel would lead to a faster impact of such a problem; hydrogen fueling of a subsonic cargo aircraft of given payload should be expected to increase H_2O emissions by a factor of about 1.7 (Ref. 27). Less engine-power is required with hydrogen-fueled aircraft (due to decreased take-off weight for a given payload); otherwise, H_2O emissions would be higher by a factor of about 2.6 for equal engine power. Future, advanced turbofan engines are estimated at comparable ratios (Ref. 27).

Other pollutants than NO_x and H_2O can be hypothesized, for example, HO_2. However. there are no known experimental data either to support or refute concern for such pollutants.

2.3.1.7 Water emissions

Unless condensed, the water vapor formed as the major product of combustion of hydrogen is emitted in the exhaust stream. Two to three times higher water-vapor concentrations are contained in the combustion products of hydrogen as in the products of typical hydrocarbon fuels at a given equivalence ratio. This leads to substantially higher dewpoints for hydrogen combustion products than for those of hydrocarbon fuels. This fact is evidenced operationally by the denser, more persistent condensation trails observed in hydrogen fueled turbojet flight tests (Ref. 33). Figure 2-5 shows the two sets of dew points for various equivalance ratios. Figure 2-5 implies that at practical equivalence ratios, hydrogen offers more possibility for condensation of water vapor than do hydrocarbon fuels. Relative to hydrocarbon-fueled gas turbines operating at modest equivalence ratios (e.g., ≤ 0.5), hydrogen-fueling would yield 25^{o} to $30^{o}F$ higher dew points in exhaust products. Even compared with diesels or gasoline engines operating with hydrocarbons at equivalence ratios of 0.8 to 1.0, a hydrogen-fueled gas turbine at lower equivalence ratio (e.g., 0.4) would exhibit higher product-gas dew points by $10^{o}F$ or more.

There is an exception to the previously-cited higher dewpoints of hydrogen-combustion products. Hydrocarbon fuels containing sulfur and yielding SO_3 as a combustion product suffer from higher dewpoints and corrosive (H_2SO_4) condensate. Dewpoints increase with increasing fuel sulfur and with increasing excess air in the combustion products. Dewpoint elevations above those for pure water can be as high as $100^{o}F$, exceeding those cited above for hydrogen-combustion products. In such cases, use of hydrogen as a fuel can lessen condensate problems and consequent corrosion problems in breeching and ducting.

The greater possibility of water condensation in hydrogen combustion products has several implications. First, exhaust-heat recovery might seem to be more appealing than otherwise. Advantage might be taken of the greater potential for recovery of latent heat of vaporization from a larger quantity of water vapor.

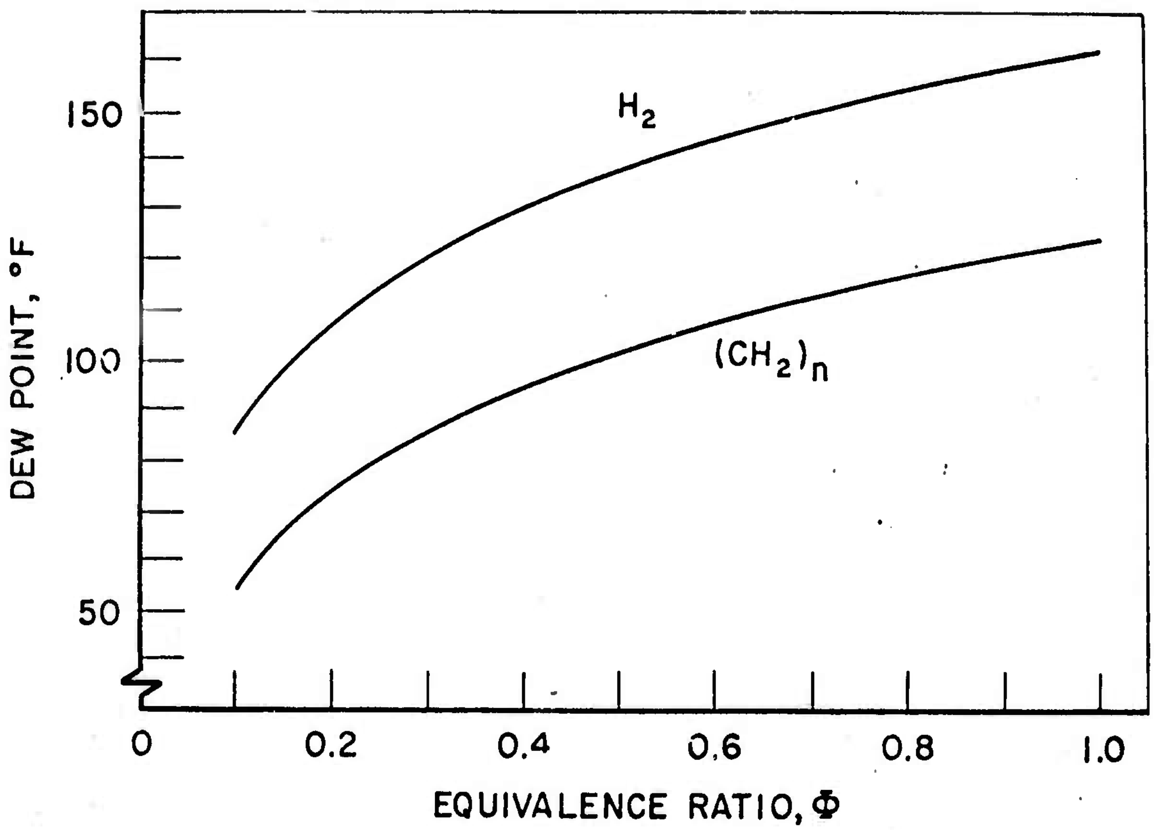

FIGURE 2-5: Exhaust-Product Dew-Point Temperatures

The greater difference between higher and lower heating values of hydrogen (18%) than of hydrocarbons (e.g., 7%) shows this difference in potential. Second, the higher dew points suggest that if, under some ambient conditions, exhaust plumes are a problem with hydrocarbon fuels, they will be worse, e.g., two to three times worse in quantity, with the burning of hydrogen fuel. Third, condensation of water in exhaust ducting might be expected to be more of a problem with hydrogen fuel than with hydrocarbon fuels (at least those of low sulfur levels).

Regarding exhaust heat recovery, it is clear from heat balance considerations that the latent heat available from condensation of exhaust-gas water is not particularly significant compared with the usual appeal of regenerating heat from hydrocarbon-fuel exhaust products. While available in greater amount from hydrogen combustion products, the latent heat of condensation is available for recovery only at rather low temperatures (dew points) and cannot effectively be used to improve power-plant efficiency. Furthermore, compared with the heat transferred in cooling products to the dew point (e.g., 10,000 to 50,000 Btu/lb of H_2), the latent heat itself is of modest proportion (e.g., 9,400 Btu/lb) and only twice as high proportionately with hydrogen as with hydrocarbon. Practically, it appears unlikely that exhaust heat recovery with hydrogen fuel would be much more attractive than it is at present with hydrocarbon fuels. The size, weight, and maintenance required of such heat exchangers has kept their use at a low level in gas-turbine power plants.

Condensed-water exhaust plumes have been observed (as aircraft contrails) to occur and persist with hydrogen fuel under conditions for which they are not to be observed with hydrocarbon fuel (Ref. 33). Adiabatic mixing calculations show that, under the conditions involved (50,000 ft altitude), contrails could be expected with hydrogen and not with hydrocarbon fuel. The products from a hydrogen gas turbine can condense upon mixing with cold dry ambient air (e.g., $-70^{\circ}F$). In contrast, the dew point of such adiabatic mixtures of products from hydrocarbon fuel is about

$20^{\circ}F$ lower and need not lead to condensation under the same ambient conditions.

Aside from "contrails", water-condensate exhaust plumes (e.g., from marine or land installations) appear unlikely to present problems with hydrogen unless they already do (in a substantial fraction of cases) with hydrocarbon fuel. The $30^{\circ}F$ to $35^{\circ}F$ difference in engine-exhaust dew points is reflected in a similar (e.g., $20^{\circ}F$) difference in the ambient temperatures required to give condensed water vapor _via_ adiabatic mixing of cold ambient air with hot exhaust products. While current operational gas-turbine experience with water-vapor condensation under cold and/or humid ambient conditions was not collected, adiabatic-mixing calculations suggest a low likelihood of this problem. This possibility bears further investigation, however, especially as regards operating conditions at sea-level ambient pressures and high relative humidities such as might be encountered in some marine applications.

Finally, corrosion problems owing to water condensation in exhaust ducting, heat exchangers, etc., are also not expected to prove a substantial problem, despite 30 to $35^{\circ}F$ higher dew points with hydrogen. Gas-turbines systems are normally subjected to both inadvertant and purposeful water ingestion, e.g., rain in aircraft applications and fresh-water flushing in marine applications. The absence of synergistic corrosive products (e.g., SO_3) with hydrogen might be expected to render condensate even less corrosive with hydrogen fuel than with hydrocarbons and comparable to "normal" water ingestion effects.

2.3.1.8 Summary

Table 2-2 summarizes the increments in efficiency and power output estimated to be derivable from the substitution of hydrogen for hydrocarbon fuels. The values also imply the possibility of substantial turbine-reliability gains (e.g., an order of magnitude) since component life or time between major overhauls can be traded off with performance. In the sum, substantial gains appear possible. However, these derive almost entirely from the potential for operating at higher turbine inlet temperatures with hydrogen (due to decreased TVR and hot-section corrosion).

The potential for gains in specific power (e.g., output work/lb of air) are greater than those for efficiency. The practicality of such gains is apparently untested. Practical evaluation would require a substantial development program, particularly (i) combustor development aimed specifically at decreasing TVR over the requisite operating range and (ii) long-term testing to provide relative data on hot-section corrosion and lifetime.

Pollutant emissions (i.e., NO_x) are seen thermodynamically to be little different from those hydrocarbon fuels except for eliminating carbon-containing pollutants. Differences attributable to non-thermodynamic influences, notably chemical kinetics, are tested in a later section (see Section 2.3.2.2, below).

TABLE 2-3: ESTIMATED INCREMENTAL PERFORMANCE CHANGES WITH HYDROGEN FUEL

CONFIGURATION	INCREMENTAL CHANGES*		
	TURBINE INLET TEMP.	THERMAL EFFICIENCY	SPECIFIC WORK
SIMPLE CYCLE (SC)	0	-3%	+5%
SC + FUEL AT 1000^{o}F	0	+6%	0%
SC + DECREASED TVR	+ 100 to 300^{o}F	+1 to 4%	+5 to 15%
SC + DECREASED CORROSION	+ 150 to 250^{o}F	+1 to 4%	+5 to 15%
SIMPLE CYCLE + ALL ABOVE	+ 250 to 550^{o}F	+5 to 11%	+15 to 35%

*as percents of baseline (hydrocarbon fuel) efficiency for simple cycle

2.3.2 Engine Components

2.3.2.1 Compressor

In terms of the basic gas-turbine cycle, hydrogen fuel should have little influence on compressor performance. Differences in combustion-product properties influence turbine operation, however, and, therefore, turbine-compressor matching. This influence on compressor operation is not expected to be large (e.g., 1% change in compressor pressure ratio in the J71-A-11 engine at a given speed, Ref. 5).

2.3.2.2 Combustor

Aside from changes in the fuel system, the major influence of fuel type is in the combustor. The combustion characteristics of hydrogen which were previously cited as distinctive can lead to major changes in combustor design and performance.

Substituted use of hydrogen in production hydrocarbon combustors has been straight forwardly accomplished with success in a number of cases (Ref's. 2,4-6,8,23,34-36), though some conventional combustor designs are apparently less easily adapted than others (Ref. 37). Combustion efficiencies as high or higher than with JP fuel have been demonstrated in conventional combustors with hydrogen (Ref's. 6,23,34-36). As a consequence of fast reaction rates, high efficiencies maintain to lower combustor pressures than with hydrocarbon fuels. This makes hydrogen unexcelled as a very-high-altitude aircraft fuel and gave impetus to early investigations of hydrogen as a turbojet fuel (Ref. 3). At present, full-load efficiencies are commonly very high with hydrocarbon fuels in conventional combustors. The potential for relative gains from hydrogen in improving full-load performance is largely in maintaining high efficiencies with concurrent savings in weight, size, pressure drop and/or complexity as well as in pollutant emissions. These gains extend to both part-load (including idle) and small-size combustors.

Volume and Weight

Maximum volumetric heat-release rates can be calculated from relative flame speeds to be approximately an order-of-magnitude larger for hydrogen than for JP-4 at the same pressure (Ref. 27). The likelihood of achieving such decreases in combustion volume appears low unless premixed combustors are used. As a low-molecular-weight gas, hydrogen, is less easily dispersed upon injection into combustor air than is higher molecular-weight hydrocarbon gas or hydrocarbon liquid. Once injected, hydrogen cannot be expected to provide notably faster turbulent mixing than with other fuels. Therefore, without premixing of hydrogen and air, the potential of the higher reaction rates of hydrogen for decreasing combustor volume can be expected to be degraded by the need for additional combustor volume to provide dispersion and mixing.

Cases for which substantial decreases in combustor size have been reported with hydrogen have typically involved different configurations than of those of the hydrocarbon combustors which were compared with, e.g., swirl-can combustors for hydrogen vs. conventional combustors for hydrocarbon. Such unconventional combustors are not unique to hydrogen, and swirl-can-type combustors which were originally devised for use with hydrogen (Ref. 38) are now being investigated for use with hydrocarbon fuels (Ref. 27) for the purpose of reducing NO_x emissions. There is apparently no direct experimental evidence that hydrogen can provide order-of-magnitude decreases in combustion volume beyond those which could be achieved in hydrocarbon combustors by configuration changes. In fact, recent estimates have suggested 15 to 50% reduction in combustor length resulting in up to 1% decrease in turbofan engine weight (Ref. 39). Still, more substantial investigation of hydrogen premixing combustors may be warranted for those applications requiring minimum combustor volume* or relatedly, requiring minimum gas-residence time in the primary reaction zone to minimize NO_x emissions. Such efforts are in progress for hydrocarbon fuels but with little emphasis on hydrogen (Ref. 40).

In summary, it appears that substantial combustor volume and weight reductions (e.g., 50%) are possible, if undemonstrated. Similarly, lower liner temperatures and attendant increases in reliability and/or TIT are apparently possible, if undemonstrated. Premixing of hydrogen and air appears to offer the most likely basis for reductions of up to an order-of-magnitude in volume, but premixed combustors are particularly subject to problems of flash-back.

NO_x Formation

The possibility of substantially reduced combustor (reaction-zone) volume with hydrogen implies the possibility of reduced

*The value of minimizing combustor volume (and weight) can be substantial in both mobile, e.g., aircraft applications, and in stationary applications. Considering the volume and/or weight penalties associated with storing hydrogen, however, one of the main values in combustor-size reduction may be simply in the shorter turbine shaft lengths it allows.

primary-zone residence times and, therefore, reduced NO_x formation rates. Calculations indicate substantial (e.g., order-of-magnitude) decreases in NO_x concentrations are possible with lean-primary-zone combustion of H_2 (Ref. 27). However, recent tests (in non-optimum combustor configurations) have shown little improvement with hydrogen (Ref. 30).

Equilibrium NO_x concentrations in hydrogen-combustion products are not notably lower than those of hydrocarbon products (assuming equal combustor outlet temperatures). Figure 2-4 shows this. Figure 2-4 shows that NO_x concentrations are not notably lower with hydrogen even if reaction kinetics are accounted for and equal, typical residence times are considered for both hydrocarbon and hydrogen combustors. The faster reaction kinetics of hydrogen, however, allow the major energy-releasing reactions to be virtually complete and NO_x formation to be purposely quenched earlier by mixing with secondary air. This results theoretically in lower NO_x concentrations at the combustor outlet. It is particularly so in comparison with liquid hydrocarbon fuels for which extra residence time in the primary reaction zone is required for vaporization. This extra benefit of hydrogen over liquid hydrocarbon fuels would, however, be largely lost if prevaporizing, premixing combustors were to be used with liquid hydrocarbon fuel. Such combustors are currently under serious investigation with hydrocarbon fuels. If developed successfully, these hydrocarbon fueled combustors may allow reduction of NO_x concentrations to low enough levels so as to render further decreases with hydrogen fuel only marginally valuable (Ref. 39).

2.3.2.3 Turbine

The major impacts of hydrogen on turbine performance are a consequence of:

(1) altered product-gas molecular weight and specific heats as well as number of moles per unit mass of air handled,

(2) the possibility of increased temperature uniformity at turbine entrance resulting in smaller tip clearances and, therefore, higher turbine efficiencies,

(3) the absence of typical contaminants in the hydrogen fuel.

The last two of these have been dealt with above. The lack of a more detailed basis for furthur consideration of these at present confines more detailed attention to the first item above.

For a given TIT, OPR, and air rate, the larger number of moles of combustion products per mass of air with hydrogen fuel ("H_2") than with hydrocarbon ("HC") implies higher axial velocities through the turbine:

$$\frac{(V_3)_{H_2}}{(V_3)_{H_2}} = \frac{\left(\dfrac{\dot{m}_{p,3}}{\rho_3 A_3}\right)_{H_2}}{\left(\dfrac{\dot{m}_p}{\rho_3 A_3}\right)_{HC}} = \frac{(\dot{m}_p/\dot{m}_a)_{H_2}}{(\dot{m}_p/\dot{m}_a)_{Hc}} \cdot \frac{\left(T_3/\mathcal{M}_3 P_3\right)_{H_2}}{\left(T_3/\mathcal{M}_3 P_3\right)_{Hc}}$$

$$= \frac{[1 + (\frac{F}{A})_{ST}\phi]_{H_2}}{[1 + (\frac{F}{A})_{ST}\phi]_{HC}} \cdot \frac{(\mathcal{M}_3)_{Hc}}{(\mathcal{M}_3)_{H_2}}$$

where:

V = axial velocity

m = mass flow rate

ρ = density

$\mathcal{M}$ = molecular weight of combustion products

$(\frac{F}{A})_{ST}$ = stoichiometric (mass) fuel/air ratio

ϕ = mass fuel/air equivalence ratio

subscripts: "3" ~ turbine inlet

"a" ~ air

"p" ~ combustion product

From ideal cycle analysis (for TIT=3000 R, OPR=20), approximate numbers can be applied:

$$\left.\frac{(V_3)_{H_2}}{(V_3)_{HC}}\right|_{\substack{3000\ R \\ OPR=20}} \cong \frac{1 + (.029)(0.4)}{1 + (.066)(0.45)} \quad \frac{28.8}{26.9} = (.982)(1.07)$$

$$= 1.05$$

Thus, slightly higher turbine speed is required to match blade angles considering the higher axial velocities encountered with hydrogen fuel.

Because of the lower molecular weight of hydrogen combustion products, acoustic velocity ("a") is also higher, giving less difference in turbine Mach numbers ("M") than in axial velocity:

$$\frac{(M_3)_{H_2}}{(M_3)_{HC}} = \frac{(V_3/a_3)_{H_2}}{(V_3/a_3)_{HC}} = \frac{(V_3)_{H_2}}{(V_3)_{HC}} \quad \frac{(M_3)_{H_2}}{(M_3)_{HC}}$$

or, as above,

$$\left.\frac{(M_3)_{H_2}}{(M_3)_{HC}}\right|_{\substack{3000\ R \\ OPR=20}} \cong 1.05 \quad \frac{28.9}{28.8} = 1.02$$

Slight differences in specific heat ratios (C_p/C_c) have been neglected.

At these conditions, cycle analysis shows somewhat higher specific work (per mass of air) from hydrogen fueling:

$$\left.\frac{(W_{net})_{H_2}}{(W_{net})_{HC}}\right|_{\substack{3000\ R \\ OPR=20}} \cong 1.03$$

The higher specific work with hydrogen fuel implies that less pressure drop and temperature change need occur across aircraft turbine stages driving a compressor (and followed by a turbojet nozzle):

$$(T_3 - T_4)_{H_2} = \frac{(T_3 - T_4)_{HC}}{1.03}$$

or, continuing the previous numerical example; for T_3 = 3000 R: $T_3 - T_4$ = 1500 R

$$\frac{(T_4)_{H_2}}{(T_4)_{HC}} \cong \frac{1545}{1500} = 1.03$$

Therefore, the compressor turbine-outlet Mach number becomes

$$\frac{(M_4)_{H_2}}{(M_4)_{Hc}} = \frac{(M_3)_{H_2}}{(M_3)_{Hc}} \quad \frac{(T_4)_{Hc}}{(T_4)_{H_2}} = \frac{1.02}{1.03} \cong 1.00$$

In other numerical cases (Ref. 16), this ratio can decrease below unity. Thus, turbine air flow per unit area and theoretical power or thrust are increased slightly (e.g., 1%) adding to the effect of specific-power increase without worsening choking problems in the last turbine stage(s).

On this basis, a given gas turbine could be expected to operate at higher speed, specific power and possibly air rate for a given turbine-inlet temperature. The same phenomena as lead to those tendencies, however, also imply interstage and turbine/compressor mismatching. The actual performance of a given power plant with hydrogen fuel depends on the balance between the cited tendencies toward improved performance and the losses incurred due to these mismatches. Thus, tests of existing gas turbines without modification to the turbine have shown similar (Ref's. 4,23) or modestly better (Ref's. 5,6) performance. Since more flexibility is available to adjust for mismatches, existing turbojets or aircraft-derivative(free power-turbine) engines provide more potential for modest performance improvement with hydrogen than do single-shaft industrial turbines. Optimal performance would require new, slightly-modified blading, a step which has apparently never been taken in conjunction with prior tests of hydrogen-fueled gas turbines.

2.3.3 Gas-Turbine Subsystems

In addition to the basic gas-turbine engine and its components, the remaining powerplant subsystems which might be impacted by the use of hydrogen fuel are:

(1) Fuel-supply system
(2) Control system
(3) Lubrication system
(4) Turbine-cooling system
(5) Ignition system
(6) Safety system

Of these, the first three are coupled in present gas-turbines through the fact that liquid hydrocarbon fuel is commonly used as a hydraulic fluid and coolant in the control and lubrication systems respectively. However, only modification of the turbine-cooling system shows promise for cycle-performance improvement using hydrogen fuel. Hence, only the turbine-cooling system is considered at this point, the remaining systems being considered as potential problem areas only.

2.3.3.1 Turbine cooling

The current practice of air-cooling turbine stator and rotor blades as well as the turbine disc affords the opportunity to operate at higher TIT and pressure ratio with resulting gains in power output and efficiency.

Cooling-air requirements are associated with an appreciable size effect, small turbines requiring a relatively higher percentage of cooling air for the same TIT (Ref. 22). Furthermore, the detrimental effect of cooling air on gas-turbine thermal efficiency is particularly apparent at part load. This is one of the reasons that air cooling is not utilized in automotive gas turbines and that, instead, research is aimed at raising TIT's by using ceramics for the gas-turbine hot section (e.g., ARPA contracts with Westinghouse and Ford).

Gaseous-hydrogen fuel offers some potential for improving on conventional air-cooling; cryogenic hydrogen offers more; the cryogenic heat-sink can be used for chilling the cooling air which is conventionally bled at elevated temperature from the gas-turbine compressor.

Estimates were made of the effects of using LH_2 to chill turbine-cooling air. Two possibilities were allowed for:

(i) decreasing cooling-air quantity without change in TIT, and

(ii) increasing allowable TIT without change in cooling-air quantity.

A constant turbine-blade skin temperature was assumed (1600F) but with blade design altered to allow for increased thermal gradients and shock. The required percent of compressor air (CA) bled-off for cooling is plotted in Figure 2-6 versus the difference between turbine inlet temperature (TIT) and cooling-air temperature (T_{CA}) for several TIT's.

Figure 2-6 can be interpreted in terms of either alternative (i) or (ii), above. Moving to the right and downward along a line of constant TIT shows the decrease in cooling-air requirement as T_{CA} is decreased (increasing TIT - T_{CA}). For example, at TIT = 2200, decreasing T_{CA} from 1000F to 300F allows a decrease in cooling air from 2.75% to 1.4%, a decrease by 47%*. Alternatively, horizontal displacement from one TIT curve to another shows the decrease in cooling-air temperature (increase in TIT - T_{CA}) which will allow the specified TIT increase. For example, 2.75% cooling air allows a TIT of 2200F with a T_{CA} of 1000F but a TIT of 2750F with a T_{CA} of about 450F, a substantial increase of 550F in TIT. Considering the second alternative, the 550F increase in TIT translates into a cycle-efficiency increase of about 7% or, alternatively, into a specific power increase of about 30% (for a typical, high OPR, e.g., 20/1 and high efficiency index, e.g., $\eta_T \eta_C = 0.85$).

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*Halving cooling-air requirements has been demonstrated (Ref. 41 and can be calculated as thermally feasible with LH_2 heating to ambient temperature (see Section 2.4.5, below).

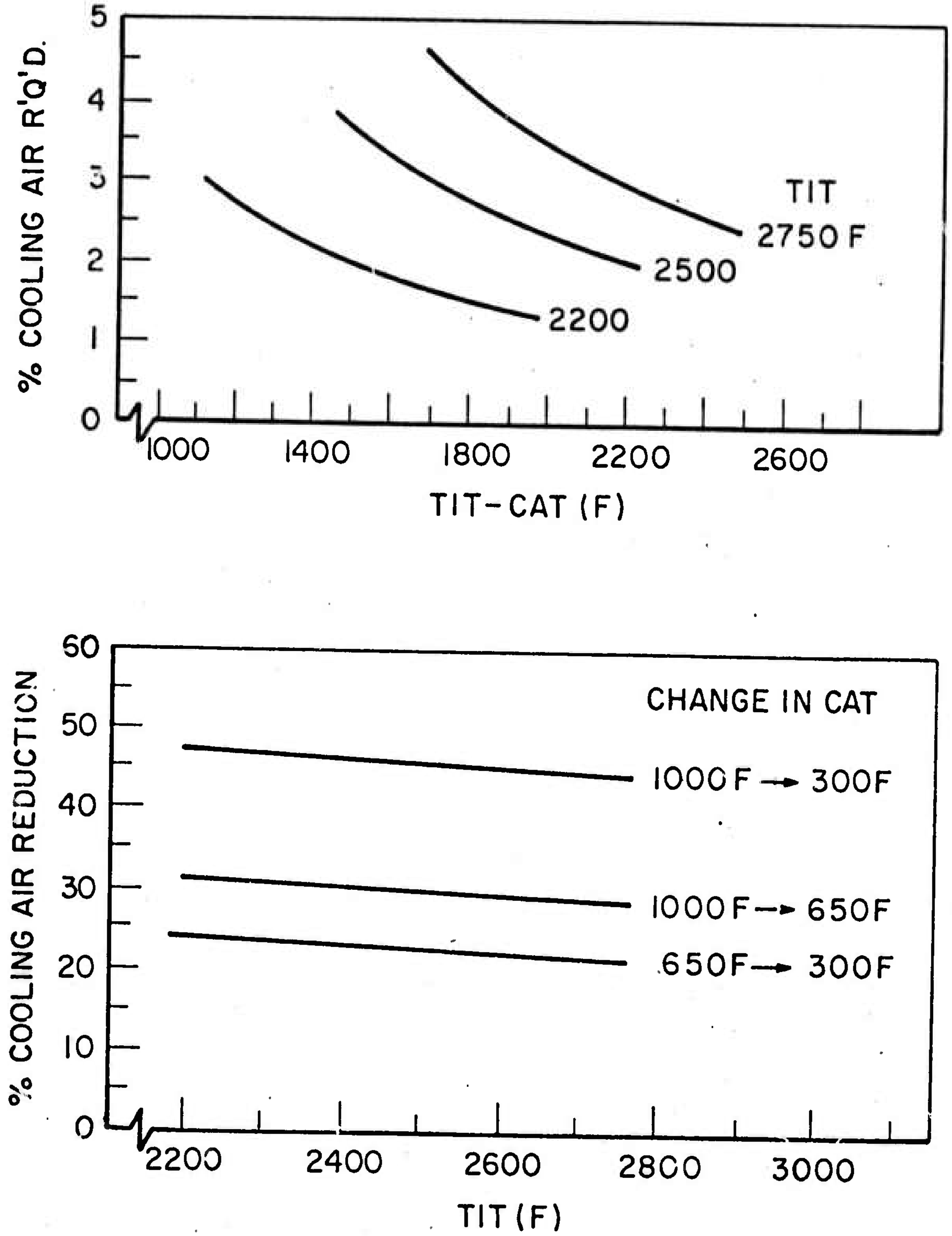

FIGURE 2-6: Cooling-Air Temperature Effects

2.3.4 Summary of Estimated Efficiency and Specific Power Increments with Hydrogen

Estimated increments in gas-turbine efficiency and specific power which might derive from substitution of gaseous hydrogen for liquid hydrocarbon fuel have been summarized in Table 2-2, above. Figures 2-7 and 2-8 add to these values the incremental performance changes which might be attributed to the use of LH_2 as a cooling-air heat-sink in a medium or large-size gas turbine. Small gas turbines typically use proportionately more cooling air (Ref. 22) and would show somewhat greater benefits of chilling.

The range shown in Figure 2-7 for efficiency increments (0% to 20%) is somewhat arbitrary since detailed provision is not made for variations in unchilled cooling-air (compressor-air) temperature as a function of overall pressure ratio which, optimally, is a function of TIT. The lower end of the range given is meant to reflect the fact that uncooled turbine cycles would not benefit from cooling-air effects. The upper end of the range shown represents what might be expected for a reasonable cooling air fraction (2.75%) and an original TIT of about 2200F (raised to about 2750F by chilling of cooling air from 1000F to 300F). While higher original TIT's (and higher original cooling-air fractions) might show larger increments in TIT (with chilling), these would not necessarily correspond with increased efficiency increments because of the well-known decreasing sensitivity of efficiency to TIT increments as TIT increases.

The specific work increments due to cooling-air chilling shown in Figure 2-8 are based on the same reasoning as the efficiency increments just discussed. However, since maximum specific work (at optimum pressure ratios) rises more linearly with TIT than does efficiency, specific work increments at higher TIT than that assumed (2200F) could be somewhat higher than those shown in the figure.

A detailed parametric study and optimization for LH_2 chilling effects has not apparently been carried out (Ref. 40) and was not included in the presently-reported work. The present lack of such

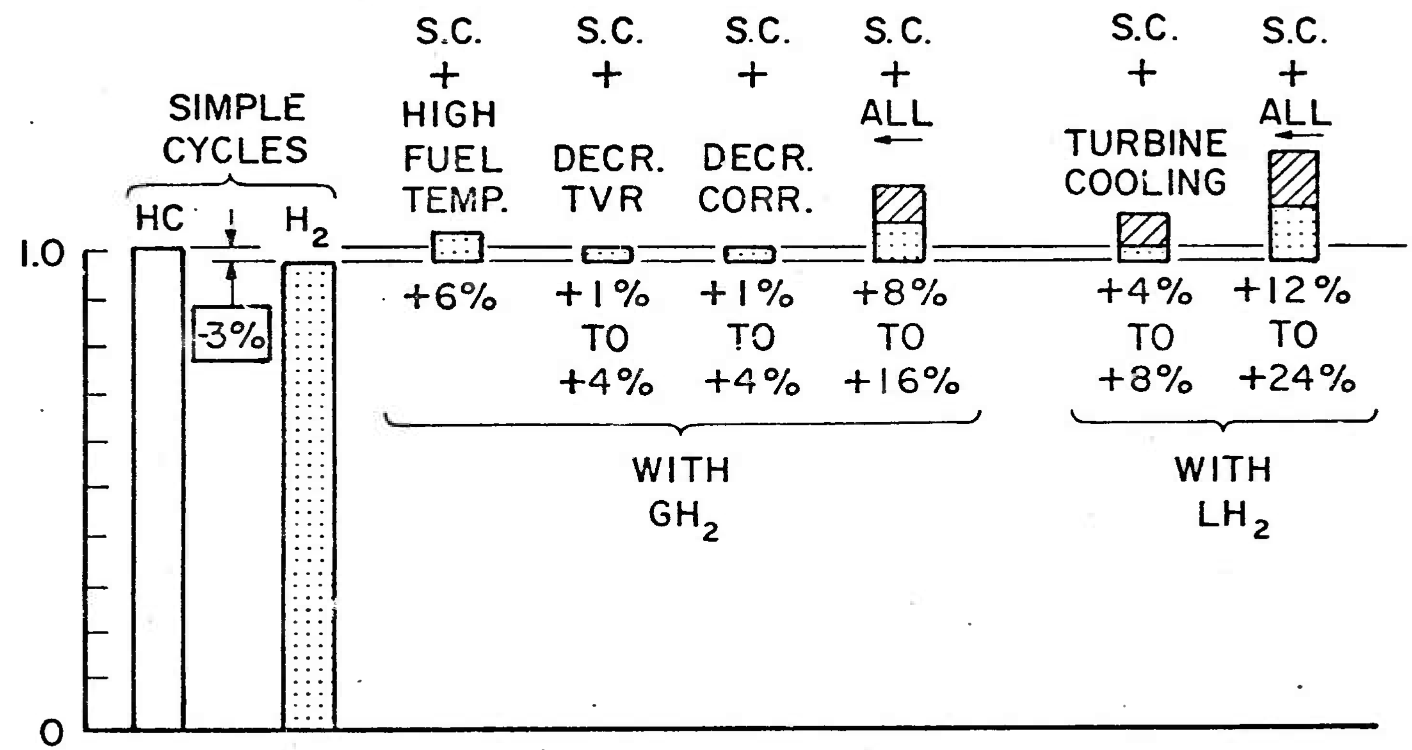

FIGURE 2-7: Estimated Incremental Efficiency Changes with Hydrogen Fuel

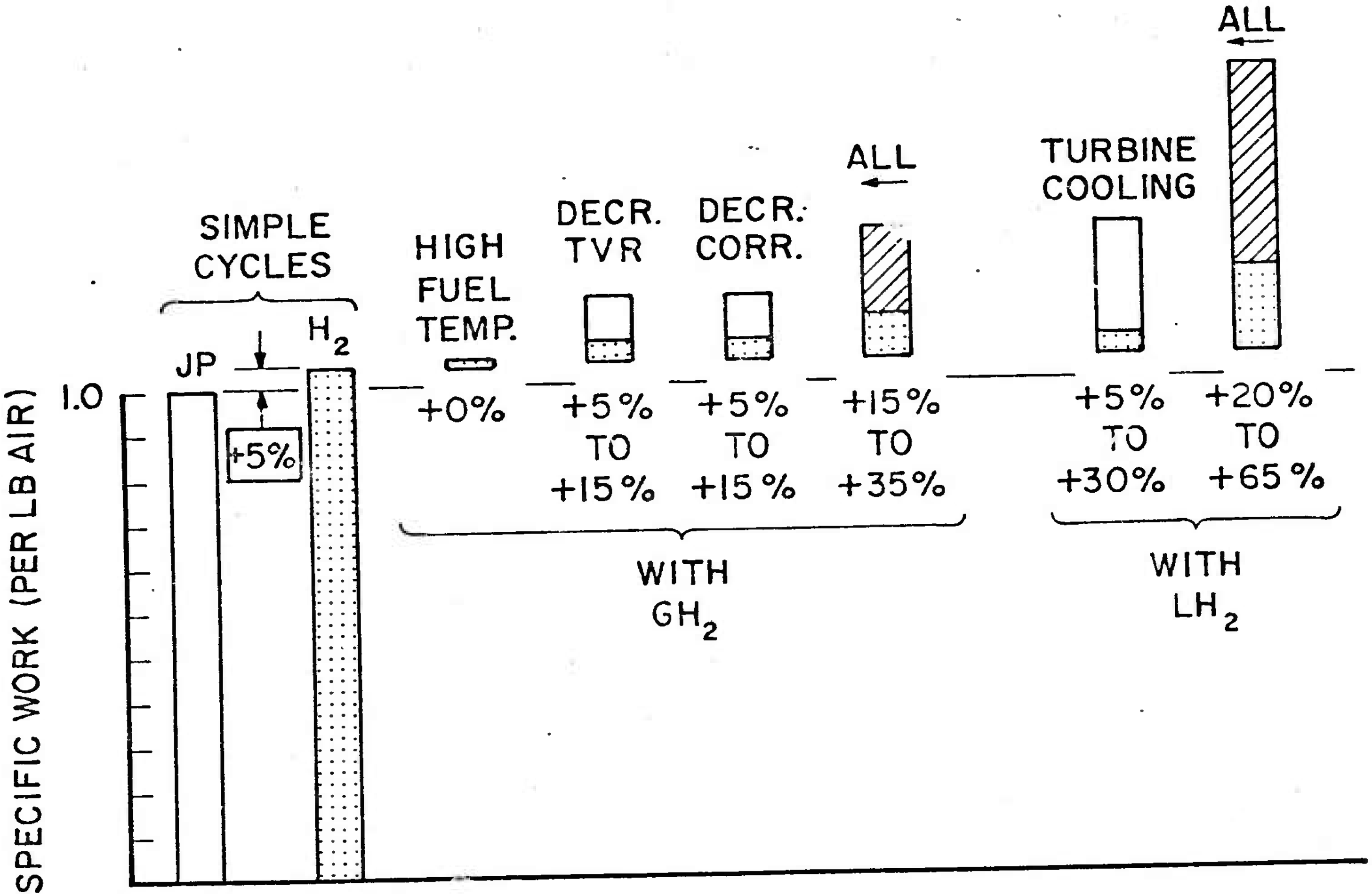

FIGURE 2-8: Estimated Incremental Changes in Specific Work with Hydrogen Fuel

information is a problem; the potential of turbine-cooling gains for offseting the various disadvantages of hydrogen fuel systems is not well defined in this regard.

2.4 - IMPACT OF HYDROGEN FUEL - PROBLEMS AND CRITICAL AREAS

2.4.1 Introduction

Normal conversior of existing gas turbines to hydrogen fuel was accomplished nearly 20 years ago (Ref's. 1,23); a hydrogen-fueled turbine provided cruise-power in a flight demonstration during 1957 only five months after program start-up. More recently, preliminary design of a hydrogen-only space-shuttle engine was completed (Ref. 26) as were the preliminary propulsion-systems specifications for both subsonic and supersonic cargo aircraft (Ref's. 20,41). Notwithstanding a long history and a multitude of efforts, several key facts hold:

(i) No aircraft is known to have completed any mission powered only by hydrogen,

(ii) No single gas-turbine converted to hydrogen is known to have undergone long-life tests of more than 20 hrs. duration,

(iii) No complete, conventional gas-turbine powerplant with hydrogen fueling has been designed and fabricated or converted and tested in a fully operational mode.

Prior gas-turbine experience with gaseous hydrocarbon fueling (e.g., natural gas) is useful in large measure when considering gaseous hydrogen as a fuel, indicating little difficulty except perhaps with hydrogen embrittlement. However, liquid hydrocarbon experience provides little utility when considering liquid (cryogenic) hydrogen fueling. Experience with cryogenic liquified hydrocarbon (e.g., LNG) may be of some value in considering LH_2 fueling problems, but it is apparent that these two fuel types, while both cryogenic, introduce substantially different problems (e.g., in fuel-system design; Ref. 49).

Despite widespread claims as to the utter simplicity of converting to the operational use of hydrogen, it is apparent for LH_2 fueling that:

(i) Major development and design efforts still remain before long-life, hydrogen-fueled gas turbines of conventional mark could be considered for practical purposes and,

(ii) achievement of the potential advantages of hydrogen in gas-turbines is either partially out-of-reach or requires substantial R & D efforts.

The complexity of Pratt-and-Whitney's proposed conversion of a military turbo-fan to space-shuttle applications is one example of these obstacles to LH_2 fueling.

The recent paper-study by Pratt & Whitney of a hydrogen-fueled-space shuttle engine (Ref. 26) demonstrates the complexity of the design, manufacturing, and operational problems with hydrogen, particularly as regards practical fuel handling, control, vaporization, and safety. Conversion of an F-401 turbo-fan was seen to require:

(1) an entirely new electronic control system,

(2) a variable-area cavitating venturi for fuel system stability,

(3) a two-stage centrifugal fuel pump driven by a new air turbine,

(4) a new air turbine to provide auxiliary mechanical power (normally fuel-pressure hydraulic),

(5) oil cooling by air (instead of fuel).

Without combustor redesign and consequent weight savings, the power-plant weight saving attributable to hydrogen use was estimated at about 3%. Fuel-pump stall was not eliminated at idle power, and engine acceleration and deceleration times were extended significantly (e.g., doubled). Fuel-pump cooldown times were long, slowing starting to nearly one-minute duration, and even the short (500-hr.) design life was not considered assured by state-of-the-art pump bearings and seals. These problems were encountered even without any attempt to consider a system which made practical use of the heat-sink capacity of the cryogenic-hydrogen fuel; the design incorporated an LH_2 vaporizer into the turbojet tail-cone.

Similar obstacles and substantial development problems have been cited in a NASA perspective on LH_2-fueled gas turbines (Ref. 1), the major problems cited being cryogenic fuel pumping and safety.

The following sections focus on individual problem areas which are apparent from these and other experiences in dealing with the practical application of hydrogen as a gas-turbine fuel. Since the large fuel requirements of large-scale mobile gas turbines (marine, aircraft) appear to mandate the use of cryogenic liquid-hydrogen storage, major emphasis is given to problems in practically fueling with liquid hydrogen.

2.4.2 Combustion

As with conventional fuels, the concurrent desires for high combustor operating temperatures (for performance) and for low pollutant emissions are in conflict. Hydrogen is an "ideal" engine fuel from the combustion standpoint with promise of substantial combustor weight and size reductions beyond those possible even with gaseous hydrocarbon fuels. Combustion efficiencies are inherently high with hydrogen even under what are normally considered extreme or normally impractical conditions, e.g., very high altitudes or very small sizes (flame-quenching effects). However, it is not yet evident to what extent NOx emissions can be substantially abated by using hydrogen fuel. Despite their complexity and problems with flash-back, premixed hydrogen combustors are promising, perhaps involving catalysis as a means for decreasing peak temperatures and, therefore, NO_x formation. Design concepts such as variable geometry, lean primary zones, recirculation, etc. which are promising for decreasing NOx emissions with hydrocarbon fuels should be even more effective when applied to hydrogen combustion.

Continuing efforts at hydrocarbon-combustor development, e.g., swirl-can combustors, can be expected to be applicable to hydrogen as well. This factor along with the extensive prior experience with hydrogen combustors suggests strongly that operational hydrogen combustors should be practical and attractive with further engineering efforts. Combustor development is a critical area only in the sense that a strong impetus for hydrogen use derives from its potential for decreased pollutant emissions. Hydrogen's theoretical potential for drastically reduced NOx emissions (Ref. 27) has not been demonstrated. This represents a weak link in an otherwise strong chain. Even advances in state-of-the-art of combustor design have been estimated to yield very modest NO_x improvements with hydrogen despite its theoretical potential (Ref. 27).

2.4.3 Safety

The practical necessity of designing and operating power-plants to eliminate fires and accidents is a critical area for hydrogen development. Much discussion has been directed at attempting to establish hydrogen convincingly as a safe fuel owing to some of its combustion and physical properties. Operating experience has, in fact, led to equipment and procedures which let hydrogen be considered as safe or safer than LNG/NG among some experienced users (Ref. 40). However, this experience has been at a limited number of sites under well-controlled conditions. For example, most current and prior "routine" uses of hydrogen as a fuel have involved open, unenclosed situations and highly trained personnel.

New uses, such as in aircraft fueling, demand further safety testing <u>specific to the circumstance of interest</u>. For example, Lockheed Aircraft, which has been very active in considering H_2-fueled aircraft designs, has been led to recommend testing in order to define better the specific problems of hydrogen safety in aircraft fueling (Ref. 41).

It is significant that few accidents have occurred and that no fatality is known to be attributable to hydrogen use. This fact appears favorable; however, the result is that information which defines what practices are unsafe is missing. Safe extension of LH_2/GH_2 to more general use (particularly in enclosed spaces) will require tight designs and procedures. However, additional empirical data are necessary in order to better define what is unsafe in operational terms for a variety of common circumstances. Since it is impractical to design and operate for absolute safety, hazards must be viewed in relative terms. This requires a definition of conditions of marginal safety and of what circumstances constitute inacceptably low factors-of-safety.

Qualitative and quantitative standards and guidelines for safe hydrogen use have been assembled and even codified (Ref's. 42-45). Quantitative data in some significant areas are apparently lacking, however. For example, the tendency of leaking hydrogen to self-ignite or, in closed spaces, to detonate is not apparently well-characterized in operationally usable terms. Particularly relevant to gas-turbine operation is the fact that the analogue of FAA ventilation-rate requirements for conventional fuel does not apparently exist for hydrogen (Ref. 40). Recommendations concerning ventilation rates are available (e.g., Ref. 45), and these concur with practice in various facilities, i.e., from 1/3 to 1 air changes per minute when high-pressure hydrogen is present. Some recommendations are also based on estimates of likely leakage quantities, e.g., 1000 ft^3/min. of ventilation per quart of LH_2 or its equivalent (Ref. 45).

The fact that leaks of high-pressure hydrogen (e.g., >2000 psi) typically self-ignite (Ref. 40,46) may be significant in some gas-turbine applications. In any event, the causes and possible prevention of this phenomenon are not apparently well understood. Static electricity has been seen as the cause of self-ignition in some quarters (Ref. 47) but no definitive information on the self-ignition phenomenon was located.

It is indisputable that the safe procedures and designs used today with hydrocarbon fuels are the result of a long evolution and that today's safe practices are in large measure specific to hydrocarbon fuels. These facts strongly dictate a staging of increased hydrogen use which gradually subjects hydrogen to less and less well-controlled circumstances as experience and purposeful safety experiments allow.

In the case of gas-turbine powerplants, significant safety provisions are:

(i) Leak prevention (particularly to preclude contaminant freezing in cryogenic lines, induction of air or oxygen and self-ignition of out-leaking hydrogen from high supply pressures),

(ii) Elimination of stray ignition sources (such as static electricity and induced ignition-system voltages),

(iii) Purging of equipment with helium (or other inerts in gaseous systems),

(iv) Maintenance of positive H_2-pressure in lines and vessels (e.g., 2-5 psig),

(v) Pressure-block systems at critical points,

(vi) Remote hydrogen-supply shut-off and "block-and-bleed" supply-line valving,

(vii) Monitoring (to detect combustible mixtures),

(viii) Control of vapor-pressure and venting, and, perhaps most important,

(ix) Adequate ventilation of enclosed areas.

The first two provisions and the last appear especially critical for military applications of hydrogen; the well-controlled situations which have led to safe use of hydrogen industrially and by NASA are apparently quite different from those implied by, for example, a naval-combat situation.

2.4.4 Control System

Basic requirements for gas-turbine control systems include stability and fast transient response, overspeed and overtemperature protection, and fuel-flow scheduling. Control-system problems with hydrogen can be broken into categories depending on whether gaseous or liquid hydrogen is involved.

Gaseous hydrogen, [illegible]e gaseous hydrocarbon fuels, requires larger piping, manifolds, etc. to accommodate the relatively large volumetric flow rates required of a gaseous rather than liquid fuel. Associated control problems can arise from the gas-"cushioning" effect, i.e., the appreciable time which may be required for the control system to adjust pressure and flow rate to a new level in a gaseous supply system. Excessive time lag precludes effective overspeed and overtemperature protection and fast transient response. Furthermore, gaseous systems (hydrocarbon or hydrogen) do not provide the commonly-used convenience of hydromechanical actuation using liquid fuel as the hydraulic fluid.

In liquid-hydrogen systems, on the other hand, some unique control problems arise because of the cryogenic nature of the fuel. This can lead to additional "cushioning" effects from the additional volume of (controlled) gas in a vaporizer, compressor etc. Control-stability problems have also arisen in some cases, apparently attributable to unsteady vaporization of liquid hydrogen (Ref's. 23,48,52). Such problems will doubtless demand attention in the design of future systems. However, prior experiences suggest the likelihood of successful development and design of such systems.

2.4.4.1 Gaseous hydrogen

An obvious practical impact of gaseous hydrogen on a gas-turbine control system is the unavailability of suitable liquid fuel for use as hydraulic fluid. This is not a fundamental problem since controls and actuations can be powered by compressor-bled air or an auxiliary hydraulic system.

A larger potential problem is that gas turbines supplied with gaseous hydrogen have shown decreased time-response to control, e.g., increased times for attainment of full-power starting from idle (Ref. 49), overspeed and overtemperature control, etc. Unfortunately, the number of experimental data available is small, and it is not possible to separate the contribution to such time lag on the one hand from gas expansion in supply lines and, on the other hand, time-lag in cryogenic pumps and vaporizers.

Simplistic analysis assuming turbulent, incompressible pipe flow suggests that a cushioning effect on control due to gas expansion or compression need not be more likely with hydrogen than with natural gas. Natural gas, in turn, has shown satisfactory speed control in stationary gas-turbines and in marine and aircraft gas turbines (Ref. 50). Supplying hydrogen to a given turbine at the same BTU/sec rate as with natural gas might lead to about 40% larger piping diameters to maintain the same pressure drop with hydrogen as with natural gas * For a given piping length, then, the supply-line gas volume downstream of a flow controller handling hydrogen would be about 100% larger than with natural gas. However, the heating value per unit volume of natural gas is over 200% larger than that of hydrogen at the same pressure and temperature. Therefore, less heating value would be "stored" in this gas volume with hydrogen than with natural gas at any given pressure and temperature. A given control action would, roughly speaking, bleed off or add chemical energy to either natural gas or hydrogen systems at the same rate (lower or higher than the original). Therefore, owing to its lower "stored" chemical energy, the hydrogen piping and manifold system even though larger in physical volume would be expected to respond faster. In fact, one early hydrogen experience indicated control response with hydrogen "quite similar to the response obtained with the normal configuration of the control system using JP-4...control-loop dynamics are not affected...(and) responses of the control system to throttle bursts conducted during flight tests were similarly well-behaved" (Ref. 50).

2.4.4.2 Liquid hydrogen

Control-system problems with hydrogen-fueled gas turbines are considerably complicated by the use of cryogenic, liquid hydrogen. These trade-off against the storage advantages of cryogenic hydrogen (Ref. 51).

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*This size increase is smaller if hydrogen gas is supplied colder than natural gas. This might be the case if liquid hydrogen is vaporized as fuel; in extreme cases, e.g., H_2@NBP and NG@NBP, hydrogen piping can be substantially smaller than that for natural gas.

Since high-pressure storage of cryogen is unlikely for large-scale usage (owing to high tankage weights), liquid hydrogen must either be pumped to suitable pressures for control and injection and then vaporized or vice-versa. The relative ease of pumping liquid compared with compressing gas favors the first alternative. Gas from a high-pressure vaporizer can then be pressure-regulated downstream of the vaporizer (e.g., Ref. 49) or, alternatively, high-pressure liquid can be flow-regulated upstream of the vaporizer (e.g., Ref. 26).

Controlling hydrogen gas-flow downstream of the vaporizer ("gas-side control") has the advantage of isolating the turbine combustor from the vaporizer, contributing to stable fuel-feed in the face of possible fluctuations in pressure and output-flow of the vaporizer (Ref's. 36,52). On the other hand, with this control scheme, the fuel pump's discharge state is coupled to fluctuations in vaporizer conditions. This has been anticipated as a problem area (Ref. 26). As an advantage, however, gas-side control means that vaporizer volume does not contribute to the possible gas-cushioning problem discussed above. Vaporizer volume can be significant relative to gaseous manifold and supply line volumes (e.g., approximately equal or greater as estimated from data of Ref. 26).

Controlling liquid upstream of the vaporizer ("liquid-side control") has the advantage of isolating the liquid pump from pressure fluctuations in the vaporizer. However, the likelihood of two-phase flow occurring (under some operating conditions) between controller and engine presents a problem in that the flow controller can become decoupled from the actual flow into the engine. This is a possible source of control instability. Concern is heightened by the fact that two-phase flow characteristics of hydrogen are not well-known and have been suggested as a significant problem area in hydrogen fuel-system design (Ref. 49).

It appears, then, that transients and fluctuations in hydrogen vaporizer operation are significant potential sources of control problems in liquid-hydrogen-fueled gas turbines. Most hydrogen gas-turbine tests have involved gaseous fuel. Nevertheless, some tests of a particular gas-side control scheme with liquid hydrogen and a reciprocating pump have indicated stable operation (Ref. 49) though not over extensive testing. Other experienced hydrogen users have earlier chosen different control schemes, anticipating problems with gas-side control with centrifugal liquid pumping and knowing of stability problems in earlier systems. Apparently, no complete operational hydrogen-control system has been constructed and tested with a gas turbine. This leaves a need for at least substantial engineering development and possibly unexposed fundamental problems. The latter are to be expected primarily as a consequence of either hydrogen pump and/or vaporizer operation.

2.4.5 Fuel System

The basic gas-turbine fuel system must provide controlled fuel flow from storage to the combustor injector. As a gaseous fuel, hydrogen can usefully be raised to high temperature before injection (see Section 2.3.1.2). The major component elements of the hydrogen fuel system are, then:

- storage vessel
- pump(s) or compressor(s)
- liquid vaporizer and/or gaseous-hydrogen heat exchanger
- supply lines
- manifold and injector nozzles

Problems and critical areas involving storage vessels have been considered in earlier technical reports of the current program (Ref's. 51,53). Table 2-3 summarizes problem areas likely to be encountered with other fuel-system components, and these areas are considered in more detail in subsequent sub-sections.

TABLE 2-4: FUEL-SYSTEM-COMPONENT PROBLEM AREAS

COMPONENT	PROBLEMS
LH_2 Pumps	State-of-the-art hardware development for decreased weight, increased lifetime and reliability, and for meeting fuel-flow turndown requirements.
GH_2 Compressors	Hardware development for reliability, minimization of compression-work inputs required.
LH_2 Vaporizers	Transient response, off-design (e.g., turndown) operation including energy balancing with heat source.
GH_2 Heat Exchangers	(not investigated)
Supply lines, manifolds injectors	–

2.4.5.1 LH_2 Pumping

If hydrogen is stored as cryogenic liquid, modest storage pressures, e.g., 1 to 5 atm, are indicated in order to minimize tankage weight. Combustor injection at sea-level operating conditions requires 8 to 10 atm (low OPR, industrial gas turbines) or up to 40 or 50 or more atm, (high OPR, aircraft or aircraft-derivative engines). Therefore, LH_2 pumping is required unless operation is at very high altitude (low combustor pressure). Early hydrogen gas-turbine experiments (including the only engine flight test known) were at low combustor pressures, and LH_2 pumping was not required. No complete hydrogen-supply system for a conventional gas turbine is known to have been operated. One of the major reasons for this appears to be the difficulty in providing suitable cryogenic pumping. Recent attempts to produce flight-weight liquid hydrogen pumps were not successful though they were not extensive (Ref. 49).

Requirements for LH_2 fuel pumps in virtually any gas-turbine system include:

- high turndown ratio, e.g., 20:1 to 30:1 or even 70:1 in exceptional cases (Ref. 49)
- high delivery pressures, e.g., 500 to 1000 psi
- long life, e.g., 100,000 hr.

In some applications, additional features are required:

- small size
- low weight
- short chilldown time

Unlike conventional liquid fuels, cryogenic hydrogen is not readily recirculated to storage at times of low demand; boil-off in storage can be increased excessively by recirculation of heated fuel.

Under these conditions, variable-speed reciprocating pumps have been designed and used with LH_2 gas turbines (Ref's. 23,50). So far, however, they have not proven reliable; piston and valve sealing are problems, and chilldown and weight problems appear to be inherent.

Centrifugal LH_2 pumps have also been designed and used. Notable are those routinely used for LH_2/LOX rocket-engine fueling, e.g., of the RL10 rocket. However, these centrifugal pumps are not well adapted to high turn-down requirements and are not designed for long life. No examples of long-life, high-turndown pump systems for gas-turbine use with hydrogen are known.

A two-stage LH_2 pump, derived from the RL10-rocket pump, has been preliminarily designed by Pratt & Whitney in conjunction with a proposed space-shuttle engine (Ref. 26). However, it has not been built and tested. Earlier, Pratt & Whitney did design, build, and test an LH_2 centrifugal pump for the 304 hydrogen-expander engine. However, high turndown does not seem to have been provided for this system.

In addition to difficulty in handling turndown requirements, a nagging problem with centrifugal LH_2 pumps is the difficulty in handling the two-phase flow necessarily present at start-up (Ref's. 49,51), and pre-chilling may be required. Furthermore, no long-life data are known of, only short operating times (e.g., 25 hrs) being assumed based on rocket and hydrogen-expander working experience with LH_2.

Vane-type positive-displacement pumps are potentially appealing for LH_2 gas-turbine service (Ref's. 1,26). However, internal leakage is high with LH_2, wear is an inherent problem, and no nearly satisfactory examples are known for LH_2; these pumps have been used with conventional gas-turbine fuel. Turn-down via variable displacement has not proven satisfactory with vane-type pumps, and therefore, variable-speed operation is required. A major appeal is in the weight advantage relative to reciprocating pumps.

In brief summary, liquid-hydrogen pumps which are effective for gas-turbine service have not been developed despite a variety of attempts, and such pumps represent a key engine-system component requiring substantial engineering development regardless of which type, centrifugal, vane, or piston is to be used.

2.4.5.2 GH_2 Compression

Earlier reports have indicated hardware problems with mechanical GH_2 compression both in terms of hardware reliability, size, and cost (Ref's. 51,53). Furthermore, since thermal efficiency is a particular deficiency of gas turbines relative to other power plants, any requirement for mechanical gas compression also subverts gas-turbine efficiency. Table A-2 of Section 1, Appendix A of this report indicates, as a fraction of fuel heating value, the substantial efficiency decrements which might realistically accompany GH_2 fueling if mechanical gas compression is required for attaining elevated injector pressures. Efficiency losses attributable to gas compression might be expected to range from 6 to 16 percentage points (with 60% adiabatic compression efficiency) or 4 to 8% percentage points (with 60% isothermal efficiency).

Gas-compression losses in system efficiency need not necessarily appear as losses incurred during a particular gas-turbine mission, e.g., vehicular. The operational fuel system may store high-pressure gas (or liquid), the cost of compression being charged to the original hydrogen production or supply facility rather than to the gas-turbine system itself. Still, in comparison with liquid alternative fuels, hydrogen must ultimately be billed at some point in the energy chain for the cost of gas compression.

Expenditure of mechanical work for gas compression may be "saved" in several ways. For example, a non-mechanical gas-compression scheme based on the temperature-dependent absorption/desorption characteristics of metallic hydrides is possible (Ref. 54). Such "saving" comes at the cost of substantial additional equipment to provide heating of a hydride bed and incurs the problems of hydride storage (Ref. 51). Furthermore, were gas compression not required for hydrogen fueling of gas turbines, the hydride compressor could be used in conjunction with an appropriate expander, i.e., as part of a heat engine cycle, to provide additional output work rather than simply to provide pressurized hydrogen. Thus, the work of gas compression must still be charged to the turbine as a cost, i.e., as mechanical work which otherwise could be made available for other purposes.

Only if condensed-phase hydrogen, e.g., LH_2, is available from storage, can the energy cost of gas compression be substantially eliminated as an efficiency decrement to the gas-turbine power plant. Compression of LH_2 to combustor pressures expends negligible work, e.g., less than 1% of the fuel's heating value. While injection of high-pressure cryogenic hydrogen has been demonstrated as workable (Ref. 36), it represents an unnecessary efficiency loss in that the heating value of cold fuel is less. Table 2-4 shows the specific enthalpy of hydrogen fuel streams at temperatures below ambient and indicates the consequent decrements in gas-turbine power-plant efficiency attributable to cold-fuel injection. It is apparent that pressurized LH_2 should be vaporized and superheated before injection, most likely by heat recovery from the engine-exhaust stream.

TABLE 2-5: EFFICIENCY DECREMENTS ATTRIBUTABLE TO COLD-FUEL INJECTION

Combustor Inlet Temp. (F)	$(h-h_{80F})_{\text{normal } H_2}$ (BTU/lb)	$\Delta\eta = \frac{h}{LHV}$ (%)
80 (300K)	0	0
-100 (200K)	-600	-1.2
-280 (100K)	-1140	-2.2
-423 (20K)	-1130 (para liquid)	-3.

2.4.5.3 LH_2 Vaporization

LH_2 Vaporization - Uses

Conversion of low-pressure liquid hydrogen(stored in"para"form)to gaseous hydrogen("normal"GH_2)for gas-turbine fueling requires a heat input of up to 5200BTU/lb (i.e.,up to 10% of the lower heating value of hydrogen) depending on final gas temperature. This heat-sink capacity is usable in a variety of ways directly connected with the gas turbine; e.g.,

(1) lube-oil cooling,
(2) chilling of turbine-cooling air,
(3) exhaust-gas heat recovery and/or conditioning,
(4) engine-air-flow cooling, e.g., compressor inter-cooling, as well as for other purposes, e.g.,
(5) auxiliary heat-engine operation, e.g., to recover exhaust heat,
(6) cooling, e.g., heat-shielding, cryogenic equipment,
(7) air-conditioning and refrigeration, e.g., aircraft cabin or aerodynamic heat loads.

One of the most obviously useful and earliest proposed of these is the chilling of turbine-cooling air (e.g., Ref. 48). As pointed out above (Section 2.3.3), turbine cooling by means of liquid hydrogen holds promise for extending gas-turbine performance especially as regards its weakest performance area, thermal efficiency. At present, however, it appears that no parametric trade-off studies have been made regarding optimum use of the heat-sink capacity of LH_2 for this purpose (Ref. 40).

A second significant potential use of liquid hydrogen as a heat sink is in conjunction with an auxiliary heat-engine to supplement the power output of the basic gas turbine. Only one study of this possibility is known (Ref. 55). This work focuses on such a heat engine as a means of both (1) pressurizing hydrogen (to required fuel-supply pressures) and (2) outputing additional mechanical work (to recoup some of the large work input required in the practical production of LH_2, e.g., 15,000 BTU/lb or 30% of the fuel's heating value).

A key aspect of characterizing and evaluating the potential of such uses of LH_2 is to consider the quantities of energy involved.

<u>LH_2 Vaporization - Energy Balance</u>

Table 2-5 shows some data derivable from simple energy balances which assume that the heat input required to vaporize (and superheat) LH_2 is transferred completely to:

(i) cool compressor air, or

(ii) chill bleed air for turbine cooling, or

(iii) cool exhaust products, or

(iv) liquify air.

Effects (i), (ii) and (iii) are shown as approximate temperature changes, and for (iv), the quantity of air liquifiable is shown as a fraction the air required for combustion (assuming 200 BTU/lb of air for liquefaction).The effects are calculated for several fuel/air (mass) equivalence ratios, ϕ.

Some basic facts are clear from the data of Table 2-5.

First, compressor air or exhaust products can provide ample energy for LH_2 vaporization without reliance on external heat sources. Conversely, LH_2 vaporization does not provide sufficient heat-sink capacity to allow substantial inlet air cooling or exhaust-gas conditioning unless power plants other than conventional lean-burning gas-turbines are considered.*

As a heat sink for pre-cooling or intercooling compressor air, LH_2 appears to offer little particular advantage compared with the more conventional alternative of ambient cooling-water or air as a heat sink. The heat of vaporization of LH_2 is only a small fraction of the hydrogen enthalpy change useful as a heat sink (see Table 2-5). Therefore, most of the intercooling heat exchange accomplished must be <u>via</u> a gas-gas heat exchange. While detailed analysis of such a situation was not carried out, it is doubtful that a major savings in intercooler size would result from inter-

*Non-conventional power plants burning H_2/O_2 stoichiometrically with an <u>added</u> diluent (e.g., H_2O)rather than atmospheric N_2 do show <u>approximate</u> energy balancing between LH_2 as a heat sink and the energy for liquefication of air and separation of O_2 (Ref. 13).

TABLE 2-6: ENERGY-BALANCE DATA FOR H_2-FUELED GAS TURBINE

		FINAL GH_2 TEMPERATURE			
		40R (NBP, LH_2)	280R (NBP, LOX)	80F (AMB.)	1000F
Δh for $LH_2 \rightarrow GH_2$					
BTU/lb of H_2:		190	1100	1900	5100
Δh/LHV(%)		0.4	2.1	3.6	9.9
ΔT of 100% of compressor air, ^{o}F:	ϕ=0.2	4.7 (47)	27 (150)	47 (470)	120 (1200)
(of 10% of compressor air turbine cooling)	=0.5	12 (120)	67 (6700)	100 (1100)	310 (3100)
	=1.0	23 (230)	133 (1330)	230 (2300)	260 (6200)
ΔT of exhaust products, ^{o}F:	ϕ=0.2	4.6	27	46	121
	=0.5	9.5	55	95	252
	=1.0	17	94	160	440
Fraction of combustion air liquified and separated*:	ϕ=0.2	.006	.02	.05	-
	=0.5	.014	.045	.12	-
	=1.0	.027	.09	.25	-

*@ 200 BTU/lb air

cooling with hydrogen gas. In fact, the potential safety hazard of hydrogen-to-air leakage and the possibility of air-side condensate or icing probably make compressor air intercooling less attractive with hydrogen than with conventional means.

The sub-ambient temperature of LH_2 does provide the possibility of exhaust-gas conditioning. Water vapor might be condensed, for example, to decrease infra-red detectibility of exhaust-gas plumes. However, Table 2-5 makes it clear that an additional cooling medium (e.g., sea water) would be required in order to pre-cool exhaust gas substantially before drying it with LH_2 cooling.

A second point clear from Table 2-5 is that 10% compressor-bleed air is borderline energetically as a sole heat source for LH_2 vaporization. Conversely, 10% (or less) bleed air for turbine cooling can be substantially chilled using LH_2 fuel. This provides potential for (i) decreasing the bleed air requirement (e.g., 10%) compared with current practice or (ii) chilling the same quantity currently used with a resulting potential for increased turbine cooling and increased turbine performance (TIT) (see Section 2.3.3.1).

While not shown in Table 2-5, lube-oil cooling is far less than ample as a heat source for LH_2 vaporization. Conversely, lube-oil cooling can be readily accomplished using LH_2 or vaporized GH_2, though with potential problems of sludging, oil freezing, hydrogen leakage(with resulting hazard), etc. for which appropriate heat-exchanger designs are most difficult to achieve.

Unsteady and off-design aspects of energy balancing between hydrogen demand and hot-stream flow rate and temperature are significant, particularly if maximum use is to be made of the LH_2 heat-sink capacity. Large turn-down is required of gas turbine fuel systems especially for transportation systems, e.g., 20/1 or 30/1, up to 70/1 in some extremes (Ref. 49). Varying temperatures of compressor bleed air and turbine exhaust also complicate the problem of matching enthalpy available to a GH_2 or LH_2 heat exchanger (e.g., vaporizer). Specification of a wide range of ambient inlet-air temperatures adds another potential problem in effective energy balancing.

LH_2 Vaporization - Heat Exchange

A variety of LH_2 vaporizers has been used with gas turbines as well as in other applications. Early NASA tests using LH_2 for chilling turbine-cooling air involved converted commercial fin-and-tube heat exchangers (Ref. 48, see Table 2-6). The later NASA flight-test of a hydrogen fueled B-57 (J65 engine) used a ram-air heat exchanger for LH_2 vaporization (Ref's. 52,59). More recently, a flight-design vaporizer was designed and tested in conjunction with a hydrogen fuel-system development program (Ref. 49). A unique, large scale LH_2 vaporizer was designed, constructed, and tested by Pratt & Whitney in conjunction with the 304 hydrogen expander engine program (Ref. 2). Other vaporizers both experimental and practical have been developed for non-gas-turbine applications (e.g., Ref's. 56,58).

A striking need in gas-turbine-fuel vaporizers is that of design for and operation over a substantial range of LH_2 fuel flow rates. While some vaporizer tests have been run over wide ranges of cold-stream flows (e.g., Ref's. 48,52), this has not generally been the case. This is an unfortunate information gap since large turn-down of fuel flow can apparently introduce flow instability and, therefore, control problems (Ref. 58). While fuel-system design and development can probably solve such problems, they have not apparently been fully investigated and, therefore, represent a "gray area" in need of further work before LH_2 gas-turbine fuel systems are assured of practicality.

With the exception of rocket-engine practice, many LH_2 vaporizer tests have been at subcritical pressures (<188 psia). Subcritical heat-exchanger design can apparently be based reasonably well on existing data and methods for cryogens in general and LH_2 specifically (Ref's. 59,60). LH_2 vaporizer testing at supercritical LH_2 pressure does not appear to have been extensive enough to validate or invalidate the heat-exchanger design procedures used. Heat-transfer correlations for supercritical conditions are not apparently as well advanced as those for subcritical pressures (Ref. 63). However, experience with the P & W 304 expander engine,

TABLE 2-7: EXAMPLE LH_2-VAPORIZER DATA

REF.	HEAT EXCHANGER TYPE	LH_2 PRESSURE (psia)	LH_2 FLOW (lb/hr)	LH_2-SIDE AREA (ft^2)	CORE VOLUME (ft^3)	CORE WEIGHT (lb)	ΔT_{FUEL} (F)	LH_2-SIDE FILM COEFFIC. (B/hr-ft^2-F)
48	Fin & tube, 18 tube passes, counter flow	75	100 to 650	60	3	200	150 to 200	-
52 59	Fin & tube, 1 tube pass	55	155 to 622	46	0.8	50*	<1 to 90	50 to 70
2	Cross-flow, 1 tube pass	1000	14,000	1200*	40*	500*	1500*	40*
49	Cross-flow, 2 tube passes	400	300 to 900	10	0.14	30	200 to 370	60*

*rough estimate from available data

(Ref. 2) and with the recent flight-design system (Ref. 49) suggest no unusual heat-exchanger-design problems.

Unusual construction techniques and materials are apparently not required for LH_2 vaporizers, at least for the limited operating periods tested so far (e.g., <30 hrs.). An exhaust-heat-recovery vaporizer has been tested under thermal-shock conditions without apparent thermal stress problems (Ref. 36). Under less extreme thermal-shock conditions, LH_2 transport trucks off-load using ambient-air-heated vaporizers over many chilling cycles.

The major heat-transfer design problems for steady-state operation are apparently:

(i) icing of moist air or water (or the congealing of lube oil) used as the hot stream, and,

(ii) long-term reliability prohibiting hydrogen leakage, especially into an air stream.

The later problem has led Lockheed to consider seriously a medium other than air for LH_2 cooling of turbine blading (Ref. 41).

Transient and unsteady flow in LH_2 vaporizers present a problem not unlike that described in Reference in the section on transfer-line cooldown. There are several substantial aspects to this problem.

The fast-start capability of a gas turbine may be compromised by the time required for vaporizer start-up. Except for the design problem of thermal stresses, the fast-start problem is probably not of major dimension. Unlike LH_2 transfer lines, the lines downstream of a vaporizer would presumably be sized to handle vapor. Thus, the higher-rate vaporization which could be expected during vaporizer childown should not "starve" downstream fuel lines; the flash vaporization should, in fact, help preclude start-up fuel-supply problems. While starving of the flow system can occur upstream of the vaporizer (Ref. 52), this can be handled by appropriate sizing of the LH_2 supply lines upstream. There is no experience known which indicates vaporizer start-up as a problem, though in most test experiences the gas-turbine test modes probably would not have allowed such a problem to be encountered and/or isolated.

Rapid vaporizer response to changing fuel-flow demand is another potential problem-area.

As with start-up, there is little reason to expect demands for increased fuel flow to be troublesome. Prior tests have indicated that throttle bursts can be accomplished, but details of such tests have not been published (ref's. 23,49,50,52). Indications of transients of as much as 15 seconds in such throttle bursts suggest impracticality in some applications (e.g., helicopters). A throttle-burst transient with LH_2 of about 13 sec. has been noted with LH_2, as contrasted with a 3.5 sec. transient with LCH_4 in the same system (Ref. 50). However, the fuel supply systems in these cases have not apparently been designed for fast response; further fuel-system design studies might eliminate response to throttle bursts as a potential problem.

Control actions for decreased hydrogen flow, however, encounter inherently slower vaporizer response due to "cushioning" by the volume of vapor in the vaporizer. This problem arises acutely in overspeed protection of free-power-turbine engines. Demands for decreased flow at originally low rates (compared to design point) should be most troubled by this type of slow response. There is evidence for such an effect, though inconclusive, in the engine deceleration data of Reference 50 and in the transient response times with fuel changes as reported in Reference 23. As expected, each of these reports indicates slower deceleration than acceleration. As in the case of throttle-burst response, however, it is probable that further design and development aimed at decreasing response times may eliminate this potential problem.

LH_2 Vaporization - Hot Streams

The easiest and simplest source of heat to use for LH_2 vaporization is the turbine exhaust gas. Regenerative hydrogen rocket-nozzle cooling and experience with the hydrogen-expander engine (Ref. 2) provide substantial experience bases for using such a hot-stream for LH_2 vaporizer. Minor H_2 leakage is probably not a substantial problem in such a system and, as pointed out earlier, energy is available in excess in the exhaust stream. LH_2

could be heated to optimally-high injection temperatures in the exhaust stream, and the problem of hot-stream frosting and freezing is easily avoided. Such advantages led recent space-shuttle design studies to call for exhaust-gas heat regeneration to vaporize LH_2 fuel. A cycle-efficiency benefit accrues using this method, as discussed in Section 2.3.1.2, above. Assuming that 3800 BTU/lb is regenerated by heating to 1000 F, fuel flow may be decreased without decreasing turbine inlet temperature, and an efficiency improvement of about 6% results (relative to that for fuel supplied at ambient temperature).

Compressor bleed air is another relatively simple heat source to use for LH_2 vaporization. If the air is to be simply dumped after use, minor H_2 leakage is not a problem since the air exiting the vaporizer is dumped or at least can be diluted quickly. As in exhaust-gas-heat-recovery, bleed-air temperatures from high pressure-ratio gas turbines are high enough to yield optimally-high H_2 injection temperature. Since bleed rates are controllable, frosting and freezing can be eliminated as possibilities by design. However, a 10% bleed rate balances closely energetically with the heat input required to vaporize and heat LH_2 fuel. Therefore, quantities of bleed air greater than 10% might be required to achieve LH_2 vaporization practically. An increased need for bleed air would compromise power-plant performance gain realized by heating the hydrogen-gas fuel.

Compressor-bleed air, rather than being dumped, can be used for turbine cooling with consequent increases in allowable TIT and, therefore, in cycle efficiency (see Section 2.3.3.1, above). Problems offsetting such potential gains include hazards from H_2 leakage into the vaporizer air stream and the need for a more-complicated-than-usual bleed-air route, i.e., from compressor to vaporizer to turbine blading. The leakage hazard has apparently led to considering the introduction of a "third" medium, e.g., liquid metal, which is circulated to cool turbine blades and is cooled itself in the LH_2 vaporizer (Ref. 42). The added complication of such a system is substantial, though it may be necessary in order to provide safety of operation.

Engine air provides another possible heat-source for LH_2 vaporization. Interstage or inlet cooling of the gas-turbine compressor could be provided *via* the LH_2-fuel heat sink.

Interstage compressor-air cooling is well-known to decrease compressor size and work. Since such is already possible using ambient air or water, the only added benefit of LH_2 interstage cooling is in reducing the size and weight of conventional interstage coolers or in cooling compressor air farther. Cooling to lower temperatures with LH_2 as a heat-sink introduces the possibility of condensation or freezing of engine-air moisture which is a source of compressor blade damage. Potential gains are also offset by frictional pressure loss on the air-side of the vaporizer as well as the required additional volume and weight of the vaporizer itself.

Frosting and freezing of inlet-air humidity also is an inherent problem with inlet-air cooling, and, therefore, this does not appear as an attractive option for LH_2 vaporization unless it is carried to the extreme of inlet-air liquefaction. This opens the possibility of air separation and the use of pure (cryogenic) oxygen rather than air as an oxidizer. While such H_2/O_2 power plants have been proposed, they are not conventional, open-cycle gas turbines and, hence, are not considered further at this point.

LH_2 Vaporization - With Heat Engine

Exhaust-gas heat recovery using LH_2 as a low-temperature heat sink is appealing thermodynamically. Low-temperature LH_2 allows high cycle efficiencies from a heat engine used for this purpose. Carnot-cycle efficiencies exceed 90% for a heat-source temperature of 80F (300K).

The early Pratt & Whitney hydrogen-expander engine (Ref. 2) used such heat recovery to provide work for air compression and LH_2 pumping. However, the quantity of energy recoverable at high efficiency is modest, limited by the heat-sink capacity of LH_2 (see Table 2-3).

An open Rankine cycle could use H_2 as a working fluid for exhaust-heat recovery. Extreme cycle pressures (e.g., 100 to 300 atm) and temperatures (e.g., 1000F) would allow high cycle efficiency (e.g., 50-60%) but only modest additional shaft work (e.g., 2700 to 3000 BTU/lb or about 5% of hydrogen's lower heating value) even for GH_2 expansion to 1 atm. Expansion to 1 atm would not, however, be used practically because of the need to recompress the GH_2 for injection into the gas-turbine combustor (see Section 2.4.5.2).

If expansion is to a suitable fuel-injection pressure (e.g., 10 to 100 atm), then the shaft work available from exhaust heat recovery decreases to approximately 500 to 1200 BTU/lb or 1 to 3% of the hydrogen's heating value. This additional work output corresponds with 1 to 3 percentage points increase in gas-turbine-system efficiency or with a 3 to 10% increase in a typical gas-turbine efficiency of 30%. Such exhaust-heat recovery can be considered as a major recovery of the energy cost of the original H_2 liquefaction (e.g., 10 to 30% of ideal, 3 to 10% of actual), but is scarcely a major improvement in the gas-turbine power plant considering the additional hardware required. Such hardware is largely undeveloped at present.

It does not appear reasonable to credit such a cycle further with the difference between this work output and the work input which would be required for fuel-gas compression if such a high exhaust-pressure cycle were not used and GH_2 were compressed rather than LH_2. While the approach of Reference 56 suggests that the energy cost of gas compression can be "saved", this is not ultimately true (as considered in Section 2.4.5.2, above).

Somewhat higher quantities of energy could be recovered by using LH_2 as a heat sink for a closed Brayton cycle employing helium as a working fluid. LH_2 would be pumped to pressures suitable for injection and would absorb heat from the helium cycle. The helium-cycle work output then becomes the heat rejected (i.e., input to the hydrogen) divided by $(1-\eta_{He\ Cycle})$. This approach increases the work output relative to that obtained from the open, H_2 Rankine cycle. In the Rankine cycle, the H_2 itself is the working fluid and its enthalpy gain is the heat input to the cycle; with the He Brayton cycle, this enthalpy gain is the heat rejected.

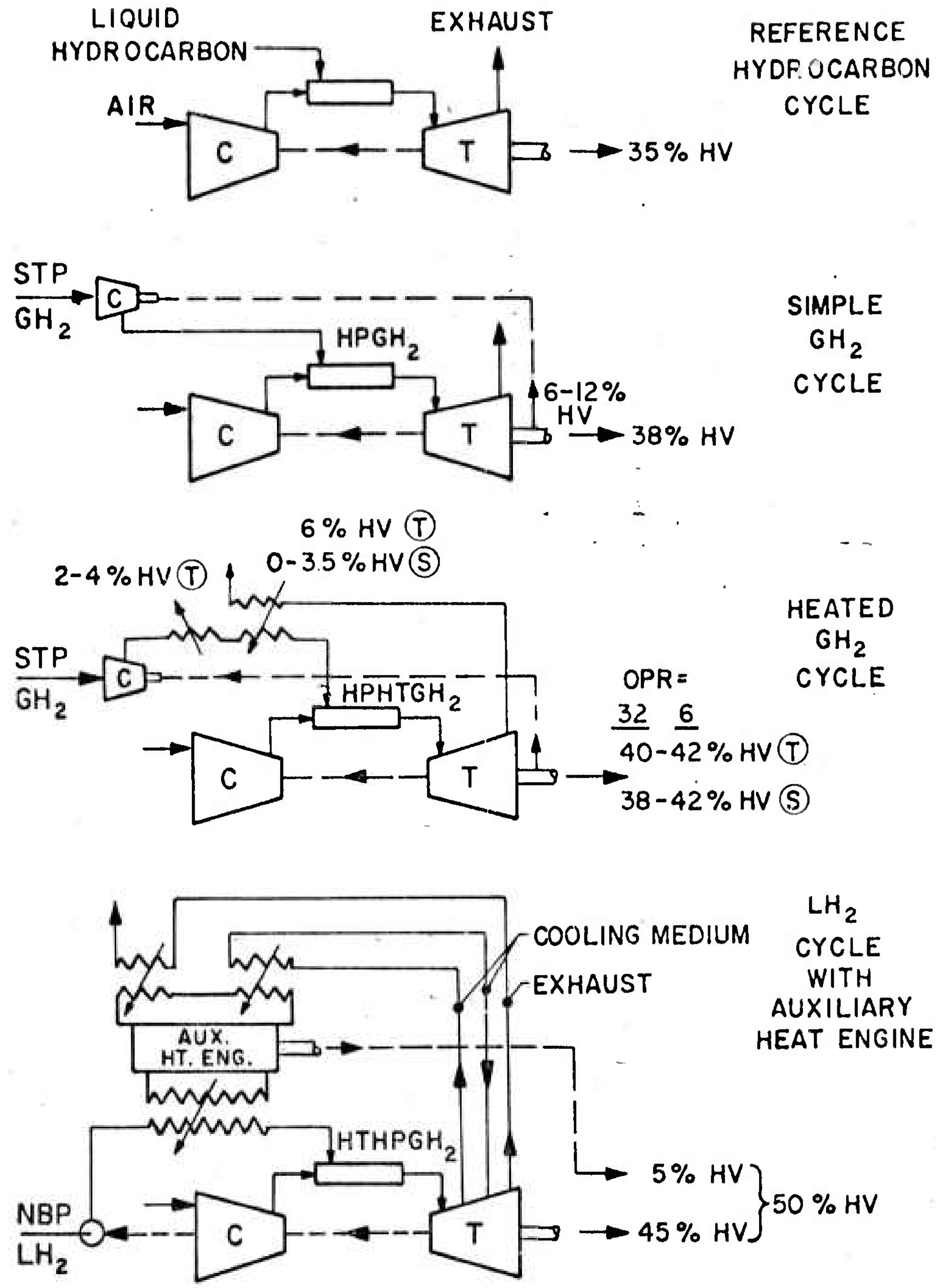
The helium cycle is temperature limited by the heat-source (exhaust-gas) temperature, but ideal cycle efficiencies of 50% or more are ideally possible. With 70% of ideal efficiency a helium Brayton cycle could yield approximately 3000 BTU/lb, i.e., 2 to 3 times that obtainable from a hydrogen Rankine cycle. This corresponds to a recovery of approximately 50% of the ideal heat of H_2 liquefaction (or about 15% of actual) or to approximately 5% of hydrogen's heating value or, therefore, to an increase of about 15% in the efficiency of a gas-turbine power plant. It is quite possible, however, that the modest gains in efficiency realizable by such a cycle would not justify the substantially increased hardware complexity required for a new, high-pressure, closed-Brayton-cycle subsystem. Evaluation depends on engineering development of currently non-existent systems.

In summary, unusually high heat-engine efficiencies are possible though undemonstrated when LH_2 is available as a heat sink. However, large energy inputs and modest efficiencies of LH_2 production more than offset these high heat-engine efficiencies, yielding low overall system efficiencies.

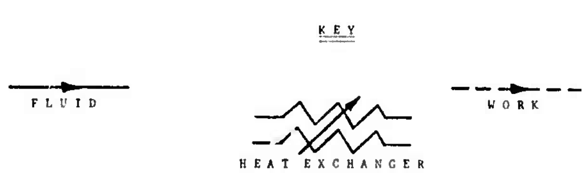
2.4.5.4 Fuel-System Cycle Alternatives

Figure 2-9 shows a rough comparison of several basic cycle alternatives including a reference, simple cycle with liquid hydrocarbon fuel. Three hydrogen-fuel alternatives are shown. Energy flows (e.g., power output) re shown as percentages of the lower heating value of the fuel. Incremental changes are to be focussed on rather than absolute values.

The first hydrogen alternative (simple cycle) shows an improvement over the assumed reference hydrocarbon-cycle efficiency (35%) by the maximum efficiency increments estimated in Figure 2-7. These increments are attributable to improved TVR and corrosion effects. It is assumed (favorable to GH_2 fueling) that all the work of GH_2 compression (adiabatic) appears as added fuel-stream



$$\text{OVERALL EFFICIENCY (INCL. LIQUEFACTION)} = \frac{50\%\,\text{HV} - 30\%\,\text{HV (LIQ.)}}{\text{HV}} = 20\%\,\text{HV}$$



STP = STANDARD TEMP. & PRESS
HP = HIGH PRESSURE
HT = HIGH TEMPERATURE
GH_2 = H_2 GAS
LH_2 = H_2 LIQUID

NBP = NORMAL BOILING POINT
T = ISOTHERMAL GH_2 COMPRESSOR*
S = ISENTROPIC GH_2 COMPRESSOR*
(* η_c = 1)
HV = FUEL HEATING VALUE (LOWER)

FIGURE 2-9: COMPARISON OF CYCLE ALTERNATIVES enthalpy and decreases the fuel requirement for a given TIT. Thus, optimistically, cycle efficiency is not influenced by GH_2 compression.

The second alternative includes in addition gains estimated to derive from regenerative heating of GH_2 fuel to 1000F; the gains depend on pressure ratio (OPR), and two GH_2 compression alternatives are shown. The figure implies a factor which has not been mentioned explicitly above, i.e., that the efficiency gains previously cited for the use of heated GH_2 are subverted if gaseous hydrogen fuel has its temperature at injection raised due to gas compression. In the event of isentropic (adiabatic) compression from STP, compressor-outlet temperatures may be so high at high OPR as to preclude substantial regeneration from the exhaust stream. With isothermal compression, on the other hand, heat corresponding to the compression work is rejected to the environment (2-4% HV). However, an even larger quantity (6% HV) can then be regenerated from the exhaust stream, yielding a net gain in cycle efficiency.

The third hydrogen alternative of Fig. 2-9 includes an auxiliary heat engine and is credited with the potential output power (and efficiency) gains cited in Section 2.3.4, above. The figure emphasizes that the use of LH_2 can make possible high energy-conversion efficiency but at the profound disadvantage of low overall efficiency due to the high practical work of liquefaction.

2.5 - SUMMARY

The fueling of conventional gas turbines with hydrogen can yield significant increases in thermal efficiency and specific power output if one or more potential fuel advantages can be developed into practice. Alternatively, these thermodynamic performance improvements can be traded off against significantly improved reliability. Possible sources of performance improvement are: (i) Regenerative heating of the fuel to elevated temperatures (e.g., $1000^{o}F$), (ii) Increased temperature uniformity at combustor outlet (allowing increased cycle temperature and decreased blade-tip clearance), (iii) Decreased hot-section corrosion problems. However, no quantitative evidence verifying such improvements was obtained. Without practical advantage gained from these, simple substitution of hydrogen for hydrocarbon fuel can be expected to show modest efficiency loss (e.g., 5%) but modest gain in specific power output (0 to 5%) compared with conventional hydrocarbon fueling.

The use of liquid (cryogenic) hydrogen (LH_2) provides an additional opportunity for additional thermodynamic performance gain through its use as a heat-sink. For example, by decreasing required quantities of compressor-bleed-air (from, e.g., 10%), LH_2 use for turbine cooling could increase efficiency and output by approximately 5%. Greater gains are achievable if future development allows blade-cooling redesign to utilize the lower cooling-air temperatures attainable <u>via</u> LH_2.

Air pollution is diminished with hydrogen fuel by elimination of CO and unburned hydrocarbon products (except those from lube oil), and pollutants might apparently be eliminated, except for NO_x emissions. NO_x emissions can be reduced substantially (e.g., 70%) from present-day levels with hydrocarbon fuels by operating gas-turbine combustors with lean primary flame zones. This gain may, however, be "empty" in light of current development toward low-emission hydrocarbon-fuel combustors. Ultimate potential (theoretically) for drastically reduced NO_x emissions with hydrogen fuel (e.g., by 2 to 3 orders of magnitude) are unproven though reasonable theoretically.

Several potential problem areas arise if simple substitution of hydrogen for hydrocarbon fuels in gas turbines is considered. Lifetime (reliability and maintenance) data are lacking, and despite favorable expectations, hydrogen fueling may raise currently hidden problems with materials compatibility (hydrogen embrittlement), sealing, safety, etc. Safety presents a currently ill-defined problem area owing to lack of operational experience with hydrogen in typical gas-turbine environments, especially marine and aircraft. This lack of experience is particularly significant in light of the detonability of a wide range of hydrogen/air mixtures in enclosed spaces. Finally, fuel-system problems are known:

with gaseous-hydrogen fuel as with LH_2, fuel pressurization is a problem area as is control-system range in the face of engine-fuel turn-down requirements. With LH_2, control-system stability and vaporizer response time are currently evident problem areas. These problem areas are detailed in the following paragraphs.

Fuel pressurization to the 20 atm or more required for gas-turbine-combustor injection is a substantial problem area. Apparently no suitable long-lifetime pumps are proven for LH_2 use. Commercially-available reciprocating pumps are special-design items and have previously proven unsatisfactory; the high fuel-flow-rate turn-down (e.g., 30/1) required for gas-turbine applications is not a common need in other LH_2 applications and, hence, suitable pumping systems are undeveloped. Centrifugal pumps have been favored in paper designs for flight-weight gas-turbine systems but are untried in long-life applications and are thought to require state-of-the-art advancement; staging is required in order to accommodate turn-down requirements. Current operational hydrogen-gas compressors are typically heavy reciprocating units with recurrent sealing problems; turbocompressors for low-density hydrogen gas are large, expensive, and currently undeveloped. In brief, substantial development is required before pressurization of hydrogen in either gas or liquid fuel can be considered routine. At the least, paper studies in this direction are required before reasonable estimates can be made of weight and size of potential compressors and pump designs. Probably hardware-prototype development is required. However, no fundamental obstacles are apparent.

Fuel-control systems used previously with gaseous hydrogen have shown little evidence of problems. However, speed of control may be a problem,for example, with respect to overspeed protection of free-power-turbine engines. Furthermore, liquid hydrogen supply systems have evidenced stability problems even over the impractically small flow-rate ranges tested. Fluctuations in two-phase flow and pressure in the LH_2 vaporizer can couple downstream with engine behavior and/or upstream with pump operation producing flow/pressure instabilities. Again, further prototype development is indicated before LH_2 control-system technology can be judged as satisfactory and problem-free.

A major additional set of problems arises if use is to be made of the heat-sink capacity of LH_2. LH_2/air heat exchangers which are safe from H_2 leakage and frosting due to humidity are crucial, and safe designs with wide flow-rate capabilities are apparently nonexistent. Off-design and transient matching of cooling load and LH_2 vaporization requirements may prove troublesome. Practical heat-exchange-system designs may be expected to degrade the gains in engine performance and reliability gains which could otherwise be attributable to LH_2-fueling unless extreme cooling techniques are used to achieve large performance gains. On the other hand, more extreme cooling-system designs are currently available but unused with hydrocarbon fuels, reflecting practical constraints on cost and reliability.

In brief, the technology for simple substitution of hydrogen as a gas-turbine fuel requires development in several areas before hydrogen-use could be considered problem-free. No fundamental obstacles have been uncovered. However, state-of-the-art advances are required before achieving the substantial performance gains which may be practical with LH_2 for turbine cooling, etc.

As regards potential applications of hydrogen gas turbines on a large scale, there are several implications. Considering industrial *vs.* vehicular applications, hydrogen has modest advantage, if any, except in aircraft applications. From a system viewpoint, aircraft applications could gain substantially from hydrogen use but primarily because of decreased fuel weight rather than from improvement in gas-turbine performance *per se*. On the other hand, considering large power-output units *vs.* small ones, small gas turbines offer more potential gain from hydrogen fueling. Small gas turbines involve long-standing problems with relatively low TIT and efficiency, high emissions, and a need for relatively more turbine cooling air than large units. As reported above, each of these performance problems could be improved with hydrogen, especially with LH_2. However, small gas turbines are probably the least likely candidates for use of LH_2 because of the added size and complexity of LH_2 fuel systems in proportion to the basic power plant size and weight.

The modest advantages of gas turbines fueled with hydrogen do not allow new, large-scale applications of strikingly high potential which otherwise are closed to hydrocarbon-fueled gas turbines. It is concluded that unless hydrogen fuel is desirable for reasons of fuel supply strategy or logistics or for reasons of its high stored heating value per unit mass, only specialized missions are likely to gain qualitatively from hydrogen use in air-breathing gas turbines.

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