

Assessment of JP-8 and DF-2 Evaporation Rate and Cetane Number Differences on a Military Diesel Engine

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ABSTRACT

The U.S. Army utilizes both world wide available diesel fuel and jet fuel (JP-8) for ground mobility applications and must maintain such fuel flexibility in order to meet mission requirements. Understanding of combustion system sensitivity to JP-8 is not well documented for such vehicle applications and thus the current knowledge base on standard diesel spray combustion must be extrapolated in order to assess fuel effects on military combustion systems. In particular, the liquid length of developed, high pressure fuel sprays is a key combustion affecting parameter that is sensitive to fuel type, the fuel delivery system, and combustion chamber thermodynamic condition. This parameter provides targeting information that is employed for assessing bulk jet mixing, cylinder pressure rise (evaporation rate), jet-wall interaction, and the formation of nitrous oxide and particulate matter.

For practical fuels it is difficult to analytically assess physical properties necessary to predict liquid length and thus well understood pure hydrocarbon fuels must act as surrogates. Typically, handbooks of such surrogate fuel thermodynamic properties are referenced to determine temperature and pressure dependence through use of an electronic library. One alternative approach to such libraries involves accurately producing curve fits of all necessary thermodynamic properties such as fuel compressibility, heat of vaporization, and density. This submission outlines such a procedure for three potential heavy hydrocarbon fuel surrogates – dodecane, tetradecane, and cetane – through utilization of a previously published liquid length model and comparison to experimental data from various sources. *The overall intent of this effort is to predict JP-8 liquid length and spray penetration in a particular military diesel engine application for assessing potential fuel-based rate of pressure rise rate issues partially through the development of a general strategy for determining evaporation surrogates for fuels similar to JP-8 and DF-2 based on a mass weighted boiling point scheme. This approach is demonstrated for three fuels - heptamethylnonane (HMN), tetradecane, and a HMN-hexadecane blend – and extrapolated for predicting JP-8 liquid length.*

INTRODUCTION

Scaling of diesel fuel-like, liquid spray penetration has been a topic of research within the engine community for nearly eighty years. As early as the 1920's, experimental work has shown that liquid penetration is limited by a number of boundary conditions and design parameters including nozzle diameter, injection pressure, charge temperature, and charge density. In particular, Miller and Beardsley[1] investigated the behavior of heavy fuel drop penetration as a function of ambient pressure and Schweitzer[2] studied the behavior of oil drops injected through a large single hole nozzle and observed such dependence that established the foundation for future scaling efforts of both vaporizing and non-vaporizing sprays. Such early efforts focused more on the gaseous portion of the spray and thus most early scaling attempts failed to account for the intact core regime.

Decades later, Ranz[3] published the aerodynamic break-up theory for predicting jet angle that has been the basis for numerous fundamental advances in the understanding of diesel-like jets[4-11]. Additionally, Levich[12] developed a two-zone theory for predicting break-up length and spray penetration that was further refined by Hiroyasu and Arai[13] and recently has been employed in various thermodynamic engine cycle simulation codes such as WAVE[14], VIPRE[15], and GT-Power[16]. Traditionally, the primary issue with such empirical models has been the lack of a significant experimental database to validate model coefficients over a variety of nozzle geometries and charge thermodynamic conditions, and furthermore, only within the last twenty years have engine representative experimental techniques and test hardware been developed to address this issue.

In particular, constant volume bombs and unique optical single cylinder engines have been the two experimental apparatuses employed for measuring liquid length. Currently, most of the published data has been generated from the former since boundary conditions are experimentally more readily attainable with a bomb versus an optical engine. The measurement techniques typically employed in such studies are Mie scattering[17-21], exciplex laser induced fluorescence[22], and

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set developed by Siebers[26]. The liquid length model is shown below:

$$L_b = \frac{b}{a} \sqrt{\frac{\rho_f}{\rho_a}} \frac{\sqrt{C_a} \cdot d}{\tan(\frac{\theta}{2})} \sqrt{\left(\frac{2}{B_s} + 1\right)^2 - 1} \quad (1)$$

where b and a are constants, ρ_f is the liquid fuel density, ρ_a is the ambient density, C_a is the area coefficient, θ is the spray formation angle, d is the orifice diameter, and B_s is the fuel to ambient gas flow rate ratio or evaporation coefficient. Furthermore, this latter constant B_s is fuel dependent and based on the iterative solution of the conservation of mass and energy applied to a rectangular control volume around a propagating fuel spray. For completeness this relationship is shown below:

$$B = \frac{Z_a(T_a, P_a - P_s) \cdot P_a \cdot M_f}{Z_f(T_s, P_s) \cdot (P_a - P_s) \cdot M_a} = \frac{h_a(T_a, P_a) - h_a(T_s, P_a - P_s)}{h_f(T_s) - h_f(T_f, P_a)} \quad (2)$$

where Z_a and Z_f are the ambient and fuel vapor compressibility, T_f is the injected liquid fuel temperature, T_a and P_a are the ambient temperature and pressure, M_f and M_a are the fuel and charge molecular weights, T_s and P_s are the fuel saturation temperature and pressure, h_a is the ambient enthalpy, and h_f is the fuel enthalpy.

FUEL PROPERTIES – Three pure hydrocarbon fuels were chosen for this study in order to encompass the range of boiling points encountered with military fuels – dodecane, tetradecane, and cetane. Saturation pressure and liquid density relationships were given by the API handbook[33] while compressibility and enthalpies were determined through accurate, piecewise curve-fits of values published in the API handbook[33]. The critical properties for each of these three fuels and their

Table 1: Surrogate Fuel Properties

Fuel	Critical Temperature (K)	Critical Pressure (bar)	Boiling Point (K)
dodecane	658	18.2	489
tetradecane	693	15.7	526
cetane	723	14.0	560
heptadecane	736	13.4	575
HMN	692	15.7	513
DF-2	NA	NA	580*
JP-8	NA	NA	496*

* 90% distillation point

corresponding boiling points are shown in table 1 along with the 90% boiling points for both military DF-2 and JP-8 fuel samples. It is apparent that some combination of these pure hydrocarbon fuels could represent the 80% to 90% distillation point of JP-8 assuming a scheme could be developed to choose the actual volume or mass fraction contributions. The formulation of this approach will be presented later in this submission and units of the following thermodynamic properties are given in Appendix 1. The saturated fuel vapor enthalpy is based on the saturation temperature and given below:

$$\begin{aligned} P_{r,s} < 0.2 & \quad h_f(T_s) = A \cdot T_{r,s} - B \\ P_{r,s} \geq 0.2 & \quad h_f(T_s) = -C \cdot T_{r,s}^2 + D \cdot T_{r,s} - E \end{aligned} \quad (3)$$

where A , B , C , D and E are fuel dependent constants given below while $T_{r,s}$ and $P_{r,s}$ are the reduced saturated temperature and pressure.

Table 2: Fuel Enthalpy Constants

Fuel	A	B	C	D	E
dodecane	1562.5	444.84	1921.9	4996.7	1978
tetradecane	1683.8	469.88	3712.2	8085.6	3229.2
cetane	1869.7	550.15	929.51	3520.3	1283.4

Next, the liquid fuel enthalpy is determined based on the ambient pressure since the control volume edge is chosen at the nozzle exit. For completeness this relationship is shown below:

$$\begin{aligned} \text{dodecane} \quad h_f(T_f, P_a) &= 1.277 \cdot P_{a,r} + 316.3 \\ \text{tetradecane} \quad h_f(T_f, P_a) &= 1.1718 \cdot P_{a,r} + 319.2 \\ \text{cetane} \quad h_f(T_f, P_a) &= 1.0429 \cdot P_{a,r} + 320.95 \end{aligned} \quad (4)$$

where $P_{a,r}$ is the reduced ambient pressure and T_f is the liquid fuel temperature. Fuel compressibility is also determined based on the fuel saturation temperature and its general relationship and associated constants are given below and in table 3.

$$Z_f(T_s, P_s) = -a \cdot T_{r,s}^3 + b \cdot T_{r,s}^2 - c \cdot T_{r,s} + d \quad (5)$$

Table 3: Fuel Compressibility Constants

Fuel	a	b	c	d
dodecane	16.85	36.104	26.425	7.5406
tetradecane	17.924	36.143	24.71	6.6857
cetane	16.587	34.594	24.531	6.869

Fuel saturation pressure is also determined from the API handbook[33] and is given below along with the associated constants presented in table 4:

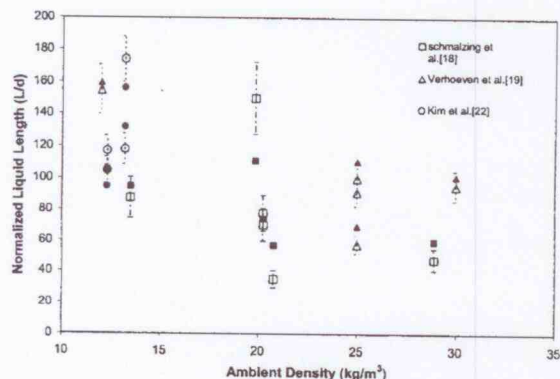


Figure 4: Comparison of the Approximate Solution with Dodecane Liquid Length Data.

fuel with a similar boiling point to JP-8 (see table 1). Data from Myong et al.[21] was only used for qualitative purposes in this submission since measurements were only reported for temperature ranges below 700K and both Sieber's calculations and the approximate solution have demonstrated an inability to accurately predict evaporation rate in this regime. Figure 4 is a collection of data from Schmalzing et al.[18], Verhoeven et al.[19], and Kim et al.[22] that shows a comparison between the approximate solution and the mean value of the measured liquid length. The 'error bars' attached to the measured data include the variance of the measurement as reported by each research team and is not attributed to the experimental measurement error associated with ambient temperature, density, and jet liquid length, i.e. it is the time history variance of each measurement. All but two of the fifteen measurements were predicted within 20% by the approximate solution and furthermore, six data points were predicted within 8% of the measurement. Overall, the approximate solution performed quite well in comparison with the associated measurements especially considering that experimental error was not included in this analysis.

SURROGATE BLENDS – Given the reasonable agreement of the approximate solution with Sieber's cetane calculations[26] and the dodecane data set leads toward the consideration of weighting surrogate fuels to simulate a real-world military fuel. For example, Siebers[26] demonstrated that heptadecane or cetane were both reasonable surrogates for DF-2 with more of an emphasis on the former versus latter. As shown in table 1, the 90% distillation point of one military DF-2 sample corresponds closely with both aforementioned surrogate candidates and furthermore, the DF-2 sample used in Sieber's study was within 1 K of the heptadecane boiling point. In this case, it is apparent that a blend of surrogate fuels isn't necessary based on purely boiling point consideration and that use of cetane as a DF-2 surrogate should theoretically under-predict liquid length due to its 3% lower boiling point versus heptadecane. Nevertheless as shown in figure 3, cetane is a good surrogate for DF-2.

JP-8, on the other hand, has a 90% distillation point that falls between dodecane and tetradecane. Based purely on a boiling point consideration, dodecane appears to be the most reasonable single fuel surrogate even though some blend of dodecane and tetradecane would provide more precise agreement with the 90% distillation point of this particular military JP-8 sample. Two approaches to attaining a satisfactory blend are considered in this submission. First, a mass weighting of the evaporation coefficient according to the specific fuel blend along with use of the actual fuel density could provide a reasonable method for predicting a real world fuel liquid length (equation 10) – this will be referred to as the Mean Evaporation Coefficient (MEC) method. Second, a simple mass weighting of the predicted liquid length, again by specific blend mass fractions, could provide a simple means to predict a real world fuel liquid length (equation 11) – this will be referred to as the Mean Liquid Length (MLL) method. Both approaches will be analyzed and discussed for three separate fuel applications.

$$B_s = \sum_{i=1}^n x_i B_i \quad T_b = \sum_{i=1}^n T_{b,i} x_i \quad \sum_{i=1}^n x_i = 1 \quad (10)$$

$$L_b = \sum_{i=1}^n x_i L_{b,i} \quad (11)$$

where B_i is the evaporation coefficient of component i , x_i is the mass fraction of component i , T_b is the fuel boiling point, $T_{b,i}$ is the boiling point of component i , and $L_{b,i}$ is the liquid length of component i .

Tetradecane – This pure hydrocarbon fuel has a boiling point almost half-way between the boiling points of dodecane and cetane, and since its thermodynamic properties are known, tetradecane offers a good test case for comparing the actual liquid length solution with that of a surrogate fuel blend. Using equation 10, the surrogate blend for tetradecane was determined to be 48% dodecane and 52% cetane. This blend was therefore employed in the MEC method utilizing B_s calculated from equations 2 and 11 to solve for the liquid length (equation 1). The same mass fractions for this blend were employed in the MLL method through equation 11.

Figure 5 is a comparison of the tetradecane predicted liquid length with the aforementioned two surrogate fuel methods. It is quite evident that the MEC method exhibits better agreement with the actual tetradecane predictions than the MLL method, especially at lower ambient densities. Though the difference in each method for this particular fuel is small, it is nevertheless important to understand that weighting the evaporation coefficient is physically more significant than a direct weighting of the predicted liquid length for each surrogate fuel component. Last, the density of tetradecane is similar to those fuels used to generate the empirical spray angle shown in equation 9 such that any

surrogate blend predicts lengths 2% - 3% higher than dodecane. Thus, either approach is reasonable as a first cut calculation of JP-8 evaporation rate under normal 'warmed up' engine operating conditions.

APPLICATION TO A MILITARY DIESEL ENGINE

Pressure rise rate in diesel engines may be correlated to the vaporized fuel mass within flammability limits at the onset of autoignition and thus liquid length behavior provides an indirect means to assess this key combustion affecting quantity. As an example, the combustion behavior of a high output, combat vehicle diesel engine was studied to assess potential issues of operating on JP-8 versus DF-2. The combat vehicle engine description is shown in table 5 and it is interesting to note that the TCM AVDS-1790 is an air and oil cooled powerplant with a low pressure fuel system that must deliver over 200 mm³ of fuel per stroke at higher loads. Due to these fuel delivery requirements the injector nozzle requires a large flow area that contributes toward potential spray over-penetration scenarios.

Table 5: Teledyne Continental Motors (TCM) AVDS-1790 Engine Specifications

Engine Parameter	Description
Number of Cylinders	12
Bore x stroke (mm)	146.1 x 146.1
Compression Ratio	14.5
Displacement (cc) ¹	2447
Coolant System Media	air and oil
Boost System	turbocharged
Charge air cooler	air-to-air with bypass valve
Injection System ²	pump-line-nozzle
Peak Injection Pressure (bar) ³	~ 650
Fuel Types ⁴	diesel or JP-8
Nozzle Geometry (mm)	10 x 0.282
Rated Speed (RPM)	2400
Maximum Power (kW)	780

¹ per cylinder

² Fuel pump delivery schedule adjusted based on fuel type.

³ Varies as a function of fuel type.

⁴ Military vehicles required to operate on world-wide variant diesel and JP-8 fuels. For this test, JP-8 cetane number was 49.

The combustion chamber is shown below in figure 10 and includes two valve unit heads and an offset piston bowl that is uniformly targeted with an offset injector. This system is subjected to both DF-2 and JP-8 and injected mass is held approximately constant at a given throttle position for both fuels through a fuel pump volume delivery adjustment. The latter fuel has shown evidence of causing erosion in the intake valve cut-out areas on both the bowl and liner corners probably due to large pressure rise rates related to over-penetration. To experimentally assess this issue, a TCM AVDS-1790 engine was instrumented for cylinder pressure and fuel

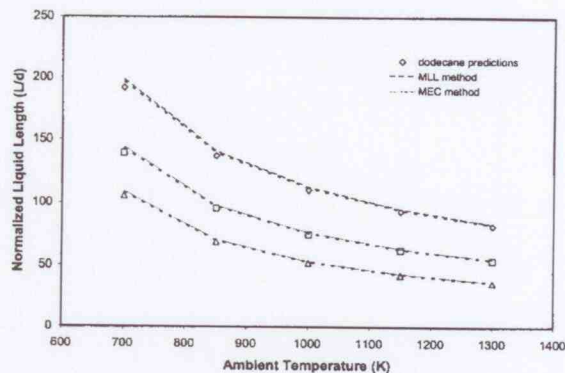


Figure 9: Comparison of Various Prediction Methods for JP-8 Liquid Length.

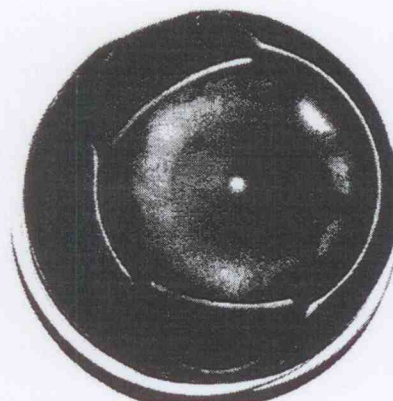


Figure 10: TCM AVDS-1790 Piston Bowl Shape.

delivery timing in one cylinder[35] and a test matrix was developed for inclusion of a few operating speeds and loads that were probable vehicle operating conditions during certain mission profiles. In particular, the test matrix encompassed engine speeds of 1800, 2000, 2200, and 2400 rpm and engine loads of 25%, 50%, 80%, and 100%, and included both JP-8 and DF-2. Furthermore, testing was completed with JP-8 that had a cetane number of 49[35] which is very high in comparison to field representative samples and thus projections will be made for a JP-8 sample with a cetane number (CN) near 40.

EVAPORATION ANALYSIS – The necessary boundary conditions measured during the TCM AVDS-1790 engine (manifold pressures and temperatures) test schedule were imported into GT-Power[16] in order to determine the initial in-cylinder thermodynamic boundary conditions at ignition, i.e. bulk temperature and density. Subsequently, Sieber's model[26] as given by equation 1 was employed to predict the liquid length of JP-8 and DF-2 by using dodecane and cetane as surrogates. The amount of evaporated fuel was determined by integrating the injected mass over the ignition delay

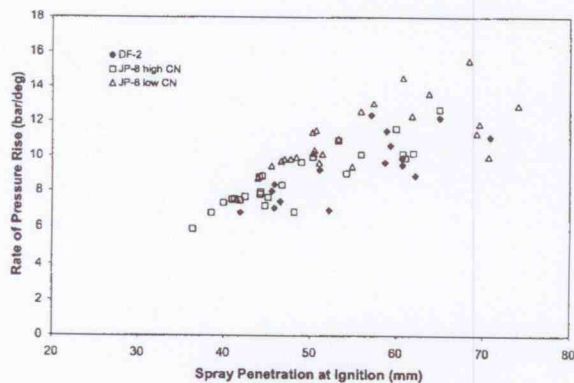


Figure 14: Influence of Cetane Number and Evaporation Rate on Predicted Cylinder Pressure Rise Rate at Various TCM AVDS-1790 Operating Conditions.

turbulent mixing and flame spread time scales at the onset of combustion. The peak fuel burning rate normally occurs within a few crank angles after ignition for most heavy-duty medium to high load operating conditions and thus the combustion chamber volume is almost constant while the cylinder pressure tends to exhibit a linear, monotonically increasing behavior during this time period. Both of these observations point toward the location of the peak premixed phase burning rate as the time and thus crankshaft location where the rate of pressure rise is a maximum assuming that ignition occurs prior to top-dead-center which is certainly the case for the various operating conditions explored in this study of the TCM AVDS-1790 diesel engine. Thus, knowledge of the peak fuel burning rate should provide a reasonable estimate of the cylinder maximum pressure rise rate and is estimated for this study based on the evaporated fuel at ignition and a representative combustion time scale. Furthermore, the maximum possible rate of pressure rise can be estimated assuming that 100% of the fuel injected during the ignition delay period is consumed in the premixed phase such that a range of probable rates of pressure rise rates may be established for a given operating condition.

As shown in figures 14 and 15, the combination of the evaporation rate and CN are major parameters that affect the projected rate of pressure rise. JP-8 inherently has a much faster evaporation rate than DF-2 and thus ignition chemistry variances between the two fuels will determine any differences in trapped fuel mass in the spray mixing layer and the spray penetration distance at ignition. Figure 15 shows that the higher CN JP-8 has comparable predicted rates of pressure rise with DF-2 while the lower CN JP-8 is anticipated to yield higher rates of pressure rise in comparison to both DF-2 and higher CN JP-8 due to its associated longer ignition delay periods and thus greater evaporated fuel mass a priori to ignition. As highlighted in figure 14, the spray penetration distance at ignition (calculated using Hiroyasu and Arai[10,13]) is only an indicator of the rate of pressure rise since evaporation rate is not explicitly

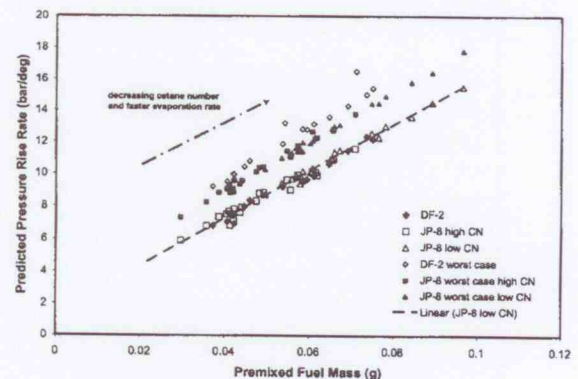


Figure 15: Comparison of the Predicted Premixed Phase Vapor Mass and Predicted Cylinder Pressure Rise Rate.

related to the gaseous phase spray formation process and instead is a strong function of in-cylinder temperature at start of injection. Furthermore, the evaporated fuel mass directly scales with rate of pressure rise since heat release rate tends to dominate the time rate change of cylinder pressure – see figure 15. Nevertheless, injection and subsequently ignition timing also impact the rate of pressure rise since the cylinder pressure at ignition and the expansion rate of the combustion chamber constrain the combustion process. In general, the expansion term contributes between 6% and 20% to the overall predicted rate of pressure rise for the operating conditions included in this TCM AVDS-1790 engine study. Last, the worst case scenario shown in figure 15 refers to the case in which all fuel injected during the ignition delay is accounted for in the shear layer premixed mass calculation. This truly is not the worst case for pressure rise rate calculations since fuel injected after ignition could theoretically be consumed during the initial premixed phase of combustion, but adequate for this simple analysis.

Overall, the use of a variable CN JP-8 and standard DF-2 presents potential issues with spray over-penetration and subsequently rate of pressure rise. According to the presented analysis, the projected rate of pressure rise will increase from 10% to 36% for a change from DF-2 to a lower CN JP-8 as dependent on the particular engine operating condition. Furthermore, a variance in JP-8 CN from 40 to 49 will result in higher rate of pressure rise ranging from 28% to 38% based on the ignition delay assumption presented earlier in this submission. Additionally, the lower CN JP-8 spray penetration distance at ignition is projected to reach the valve cut-out areas and possibly the cylinder liner under extreme conditions. The combination of higher rates of pressure rise and spray over-penetration for lower CN JP-8 may lead to an increase in thermal and mechanical stresses on the valve cut-out areas and the potential to exceed the engine peak cylinder pressure limit under high load, high speed operating conditions.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

CN: cetane number

DF-2: diesel fuel number 2

HMN: heptamethylnonane

TARDEC: Tank Automotive Research, Development, and Engineering Center

SNL: Sandia National Laboratory Combustion Research Facility

APPENDIX

Table A1: Parameters Used for Liquid Length Calculations

Reference	C_a	C_d	$\tan(\theta/2)$
Schmalzing et al. [18]	0.518	0.832	given ¹
Kim et al. [22]	0.8 ²	unknown	measured
Verhoeven et al. [19]	0.76 – 0.87 ⁴	0.696 ³	Siebers[26]
Espey and Dec[17]	0.66 – 0.75 ⁴	0.597 ³	Siebers[26]
Siebers[26]	0.8124	0.78	measured

1. Empirical relationship for spray angle given
2. Unknown and presumed by authors.
3. Calculated based on flow data.
4. Assumed c , range of 0.8 to 0.9

Units for Thermodynamic Analysis in the Fuel Properties Subheading

- Enthalpies [kJ/kg] – equations 3 and 4
- Saturation Temperature [R] and Pressure [psia] – equation 6; can convert pressure to [bar] through multiplication of equation 6 by 6894.8
- Saturation Density [lbm/ft³] – equation 7; can convert to [kg/m³] through multiplication of equation 7 by 16.018