UNIVERSAL VOLUME TO POWER DENSITY & BTE BASE NET POTENTIAL MODEL

For all positive displacement (PD) Internal Combustion Engines (ICE)

Based on reference volume of single Combustion Chamber (CC) single revolution of main shaft @ 1 Bar (absolute) atmospheric air pressure + .1 Bar.

Standard conversion factor = 1

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By: Reggie D. Huff

Abstract:

The relatively low Brake Thermal Efficiency (BTE) of all widely used Positive Displacement (PD) Internal Combustion Engines (ICE) today leaves significant space for improvement. The addiction to pistons, poppet valves & secondary crank/lever action imposes strict limitations on the common ICE, but, even in that context, much improvement is possible to a degree not possible for electric motors. At a time when ICE improvement is most important, mathematical tools that isolate BTE & Brake Volumetric Power Density (BVPD) can be combined to provide an objective means of analysis & contrast & comparison of broad categories of PD ICE designs. When these tools are designed credibly to not favor any single ICE design they become a useful aid in determining where R & D resources are best utilized for ongoing and increased ICE development. That goal was achieved where the here disclosed model as applied to the centra-fusion positive displacement (PD) multi-turbine otherwise known as the Tomahawk TX (TTX) indicates a higher magnitude potential to increase both BTE and BVPD than previously projected well beyond universally established & expected ICE limits.

Author & Bias:

The author is also the inventor, developer and substantial stake holder in one of the main subject PD ICE designs, otherwise known as the Tomahawk TX or TTX. Readers are encouraged to consider whether an element of bias may have in any way effected the final results of any portion of the analysis herein and to make adjustments or otherwise note any concerns &/or errors and freely communicate the same to the author.

Potential Brake Volumetric Power Density (BVPD):

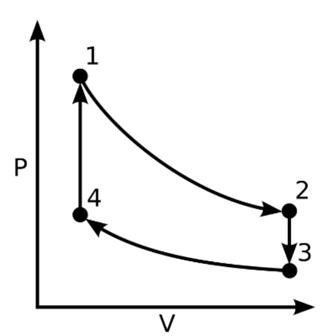
The author has developed a model for determining the base net power density potential of the most common types of PD ICE. The resulting combined equations form a mathematical model & tool that isolates the mechanical work power density potential by assuming 100% TE, or, that 100% of the potential expansion work is available before heat rejection. While there is a cause & effect of mechanical expansion work conversion to work efficiency and BTE it is not always direct and/or linear. For example, in the comparison of a Spark Ignited (SI) 2-stroke to a SI 4-stroke piston ICE, the 2-stroke has higher power density potential but less BTE potential than the 4-stroke even with the same Torque Arm Moment (TAM) profile (Fig. 4).

The potential Brake Volumetric Power Density (BVPD) is a factor of a base unit of one (1) where one (1) represents the ideal practical limited volumetric power density/**2π** ICE design. This unit is based on the contemporary and common ICEs in production &/or use. Any engine design or operating principle yielding a BVPD factor # higher than one (1) is an indication of high potential and a breakout from the constraints of contemporary ICE designs.

While much specific data exists as to the actual net BVPD (HP/L) of common engines in use the isolated mechanical potential comparison of one type of PD ICE vs another can still be useful information. This information also acts as a guide to test the equation for its theoretical approximate accuracy. Once an equation is defined and tested then new PD ICE operating principles that have not yet been reduced to practice &/or have minimal test data can be evaluated much more accurately as to their true BTE & power density potential than with simple indicated approximation, knowledge and experience based as they may be.

Single Thermodynamic Cycle:

The here disclosed BVPD potential factor (model) is based on a single thermodynamic Otto type 4-cycle occurring within a single series compression cycle with a starting absolute air pressure of 1.1 Bar supplying a single combustion chamber of the same volume (Fig. 1). It should be noted that additional cycles such as those designed to recapture waste heat energy such as that from the exhaust cycle have been shown to be able to increase final power density & in some cases BTE, by 20-40% such as the Combined Cycle Gas Turbine (CCGT).



The Tomahawk TX (TTX) is unique in this regard as the projected high base net BTE (63-72%) will greatly reduce the amount of waste heat (& noise) energy from the exhaust. However, the TTX has the capability to accomplish thermal conservation internally through a direct turbo-flux transfer and/or afterburner cycle neither of which require any additional or external plumbing (where heat is lost) or any additional operational kinetic mass (where heat is lost).

Defining The Base Theoretical Test Subject for Comparison:

The main purpose of this paper is to provide a mathematical model for comparison of a new form of engine namely a Positive Displacement (PD) centra-fusion multi-turbine or TTX to the most widely used prime mover engine, namely the Otto (4-stroke) fully reciprocating piston engine.

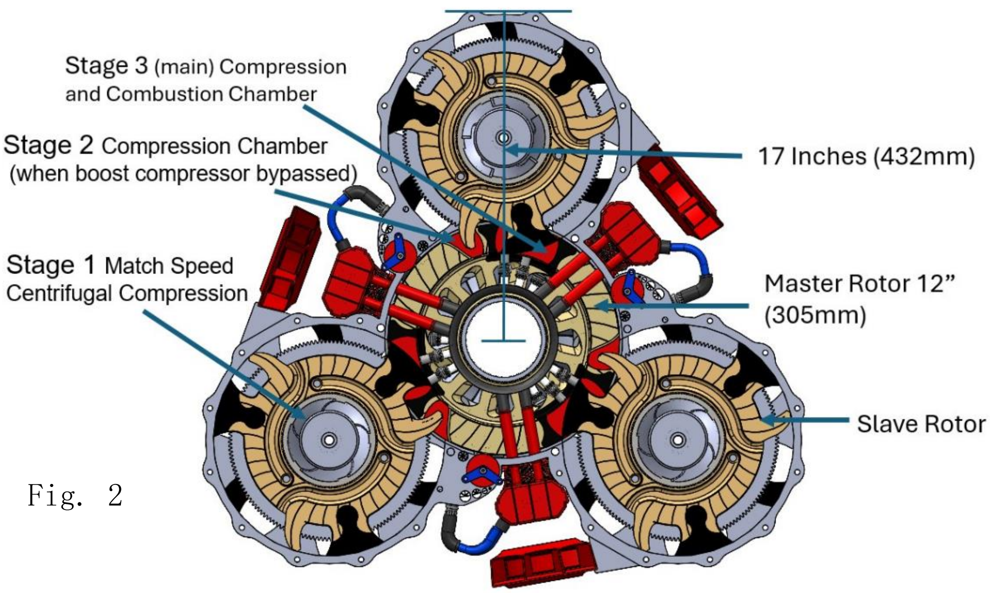


Fig. 2

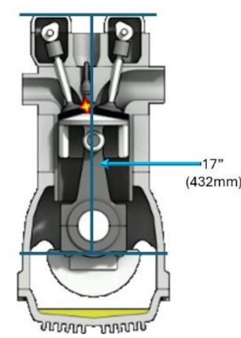


Fig. 3

In order to do this accurately, much cycle dynamic comparison information is necessary. This means that the Torque Arm Moment (TAM) and TAM profile are a critical aspect that must be quantified as accurately as is reasonably possible where no operational physical unit for at least one of the test subjects yet exists. The only way to properly compare these parameters is to consider what size of TAM will fit in a comparable sized engine. As illustrated by Fig. 2, a TTX with 12" (305mm) rotors and 1.57" (40mm) mesh yields a center power shaft to outside dimension stack height of approximately 17" (432mm). That same dimension applied to a 4-stroke piston (Fig. 3) yields a maximum expansion stroke of 4" (101.6mm). In fact, no production engine could be identified that has managed a 4" stroke within a 17" stack height not counting manifolds.

The base model assumes the same volume of air with a coefficient adiabatic expansion of 1.35 (k = 1.35) is compressed @ a ratio of 11:1 which is the knock limit for the piston ICE on pump gas. In conjunction with the TAM profile examination herein the respective knock limited CR must be corrected for the TTX to the minimum value in a range or 15:1 (See Pgs. 16 & 17 below). The model also assumes that the same fuel is injected @ the same pressure with the same inlet air temperature and the same base inlet absolute pressure of 1.1 bar. It also assumes the same initial fuel unit size and identical starting average lb pound force applied to the crank/shaft before all corrections. It assumes both are SI versions with a min two (2) sparkplugs for the piston & four (4) for the TTX. The combustion chamber material for both is essentially AL with no ceramics.

Defining A New PD Hyper-Cycle

1. Power Zone (PZ) Expansion Cycle:

Given that the purpose of this exercise is to as accurately as possible predict the BTE and BVPD potential of broad categories of engines (in this case the 4-stroke piston v PD turbine (TTX)) and not to define precise values of each, it is important to focus the analysis on the true "Power Zone" (PZ) of the respective expansion cycles.

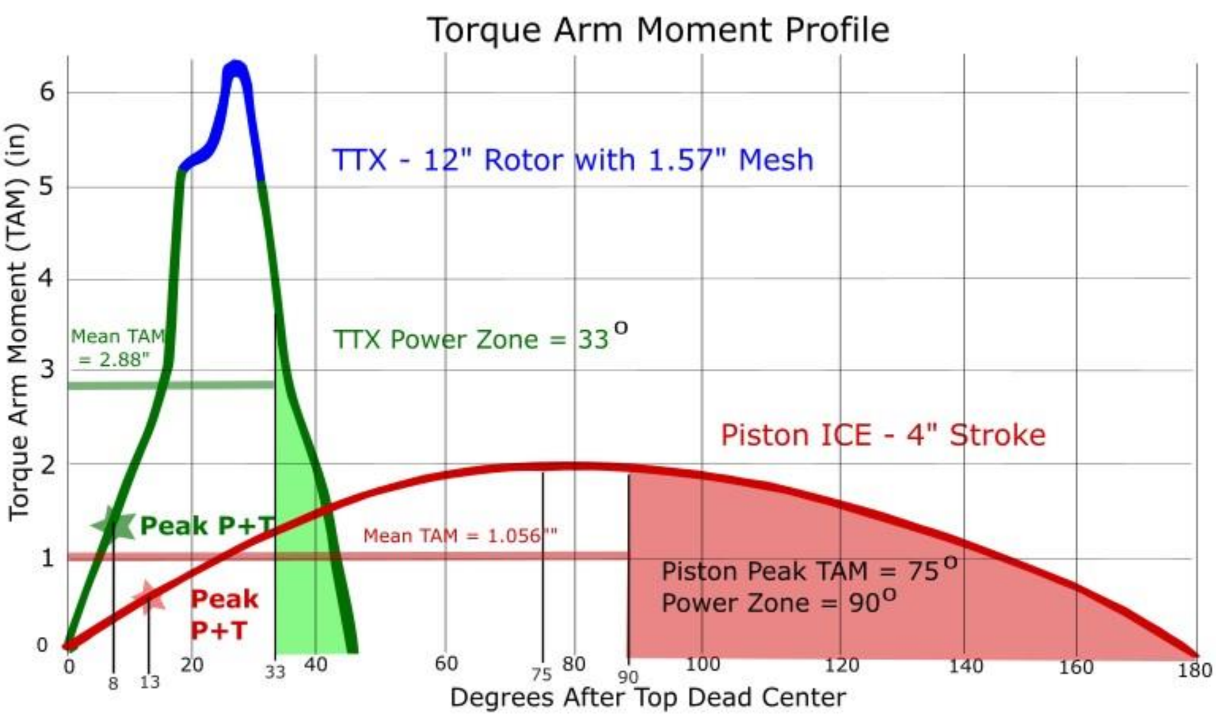


Fig. 4

Looking at the full 180° expansion cycle of the 4-stroke piston ICE starting @ TDC as illustrated in Fig. 4, The max TAM occurs approximately 15° before the half cycle which is one reason that approximately 94% of the total expansion work converted to actual work occurs in the first 90°. This defines the true PZ of the expansion cycle (Fig. 4). Where cycle timing is a critical part of any comparative and/or predictive model, not breaking the cycle down to its essential element, especially where there is such a large disparity between cycle segments, has the very real effect of corrupting the final results and frustrates the goal to keep the model within acceptable minimal margins of error. For example, comparing the full 4-stroke expansion cycle (180°) to the full TTX expansion cycle length (45°) = a 4:1 differential. But, by comparing the respective PZ cycle the ratio is reduced to 2.72:1 (Fig. 4) a number that substantially favors the 4-stroke piston, but, in fact greatly increases the model’ s accuracy.

2. **Indicated Torque for Each Expansion Cycle:**

Now that the “Power Zone” (PZ) expansion cycles have been defined and the mean TAM for each engine (Fig. 4) within the same stack height the indicated torque/**2π** for each can be calculated and compared forming a foundational starting point. We do this by applying the same mean force to the TAM profile across each PZ cycle and determine the indicated converted work in the form of torque output/**2π** before any thermodynamic, mechanical & speed related corrections are considered.

The formula for this is as follows:

$$\mathbf{P.Z_{\mathit{e}}.T_{\mathit{i}} = F \cdot \left(\bar{x}_{TAM}/12\right) \cdot \left(P_{\mathit{e}}\,\theta/2\pi\right) =}$$

$$\mathbf{1_{\mathit{TON}} \cdot (Mean\ TAM/12") \cdot (Power\ expansion\ cycle\ angle/360°) =}$$

PZ Indicated Torque - Piston = 2000lb · .088 · 90/360 = **<u>44 ft/lb</u>**

PZ Indicated Torque - TTX = 2000lb · .24 · 33/360 = **<u>44 ft/lb</u>**

It is noteworthy that given a myriad of dimensions & variables that could have been selected the chance that these parameters would produce an identical result is quite low.

Although each PZ cycle has an identical indicated geometrical and mechanical work conversion capacity each of these cycles will produce vastly different actual brake work conversion capacities as detailed herein.

First, it is a given that the amount of fuel required to produce a 2000lb force over a 33° CA° time is objectively far less than that required for a 90° CA° time over the same displaced volume. Further, both by virtue of the substantially reduced fuel load & the reduced PZ cycle time the rejected heat loss fraction must be substantially reduced compounding increased efficiency. Still further, reduced heat rejection can be one contributing factor to raising the CR knock limit which further directly increases BTE compounding the effects even further creating a true hyper-cycle (See Pgs. 16 & 17).

Even further, the reduced cycle time creates space in time to propagate more of the new hyper-cycles within a given standard unit of time (most notably **2π**) within a single combustion chamber than previously thought possible.

The combination of these factors cascade into even more favorable factors leading to a rare opportunity to combine never before obtainable levels of power density with never before achievable levels of thermal efficiency.

3. **Sealing:**

Many may observe and conclude that with a rotor spinning inside a substantially square faced housing the TTX will have the same inherent sealing issues cascading into other inherent issues, as the Wankel (& Liquid Piston & Omega 1). This is, however, a very short-sighted and rudimentary analysis that ultimately leads to a squarely incorrect conclusion.

The reasons are not limited to and include:

1. All TTX combustion chambers build the same pressure at the same time (unlike the Wankel, liquid piston & Omega 1, Etc.), thereby creating a "Jam Seal" effect that reduces the potential blowby by at least 50%.

2. Naturally high-speed cycling: The TTX cycles twice as fast as a two (2) stroke, four (4) times as fast as a four (4) stroke & six (6) times as fast as the Wankel, Liquid Piston or Omega 1, Etc. reducing the remaining blowby dimension by an additional 50-83%.

3. low heat rejection (See final #s herein, Pgs.17 & 18) combined with the most even heat distribution of any engine design allowing for tighter tolerances to be maintained more than any other ICE engine further reducing the remaining potential blowby 50-60% for a sub total of 75-93%.

4. Physical separation of chambers unlike the Wankel, Liquid Piston & Omega 1, facilitating use of high-speed surface agitators to dynamically reject &/or divert blowby & even convert a portion to positive rotational energy = - 65% (min) of remaining potential blowby.

5. Any blowby is simply recycled and reintroduced as clean (no oil contamination) intake charge or clean EGR.

Projected total blowby rejection/reduction without separate hard contact sealing apparatus compared to single piston to cylinder ICE running without high friction rings = 91-98% (higher against the Wankel). Given that about 1% blowby @ peak load & speed is acceptable for ringed piston engines, this would predict a blowby increase of 1.5-2.5% up to 6,000 rpm, an amount that might not be ideal for most ringed piston engines but would not result in failure to produce high levels of power either. However, @ speeds approaching 12,000 rpm the dynamic effects are expected to lower that number to less than the piston or .5 - 1% total blowby. *By removing the pressurized liquid lubrication acceptable blowby is increased dramatically especially since piston engines without any rings have been shown to be capable of running and producing good compression @ all speeds. When factored against the friction, heat, wear, added parts, cost & complexity of constant contact seals a blowby percentage of 15%* (6-10X the amount projected) *would be workable where oil contamination is eliminated.*

Therefore, for purposes of this analysis the only truly important consideration is whether each compared engine design can achieve the same effective knock limit static CR of at least 11:1 (4-stroke piston) & 15:1 (TTX) @ useful running speed. The three (3) stage compression cycle of the TTX (four (4) stage when the main boost compressor is not bypassed) (Fig. 2) essentially removes doubt as to its ability to achieve a static CR of 15:1 and much higher even if ultimately it has slightly higher blowby percentage than a ringed piston engine. Nonetheless, for accuracy the model adjusts for the slightly higher projected blowby by reducing the volumetric efficiency (**ηv**) of 100% to 98% @ 6,000 rpm & 99% @ 12,000 rpm.

4. **Failure Probability:**

Given this unique set of factors, allowing the TTX blowby in the range of 25% effecting 15+% (uncontaminated) EGR would only result in a detuning method reducing the otherwise extreme power density by approximately 35%. This would also create an Atkinson cycle effect, much like leaving the intake valve open longer ABDC actually resulting in higher BTE. This is an important indication because the 4 or 2 - stroke Piston, Wankel, Liquid Piston & Omega 1 (Etc.) cannot afford any added leakage let alone approaching 25% and this is especially true of the Omega 1 due to its inherent compression wasting feature. *By contrast the TTX could leak like a sieve (even on purpose) and still set new records in both BTE & BVPD as the model herein indicates.* This single fact reduces the chance of outright failure to virtually zero (0) in practical & experience-based terms.

5. Reciprocating Mass Energy Loss Factor:

Few research models address and/or isolate the thermodynamic & power density cost of reciprocating mass separate from contact & auxiliary friction. Indicated Mean Effective Pressure (IMEP) formulas tend to create a low theoretical value @ peak torque-hp rpm because they do not effectively correct for the speed efficiency factor as discussed herein. The real energy cost for reciprocating mass motion therefore tends to get aggregated into overall heat rejection.

We know this is true because if it was factored it would be counted as part of the overall friction or mechanical loss % in which case the proportional base friction load, which decreases with speed and load from 9% to 3% @ max speed/load (See Figs. 7 & 8), would be offset by the increasing reciprocating energy load which increases in the same speed/load zone exponentially from near 0 (@ around 1/2 speed) to 7% (@ full speed (6,000 rpm)). The result would be that the overall friction load percentage would level off around mid-speed/load at around 9% and slightly increase to 10% @ max speed/load, a difference of 7%.

We know that reciprocating mass has a separate and distinct energy cost from other forms of friction. This is in part due to the kinetic mass necessary to smooth over the inertial spikes generated by the reciprocating mass which approximately doubles the total kinetic mass needed which will impose a measurable energy cost to accelerate that mass. That mass also increases the overall engine mass by 12-15% which has a direct effect on the power/mass (hp/lb(kg)) density.

Where a debate appears to emerge is in trying to quantify the reciprocating mass energy cost as a constant @ a constant speed. Some propose that the reciprocal energy simply shifts back & forth from the flywheel (& balancer & CS counterweights) to the reciprocal mass and back completing a reciprocal energy feedback loop where effectively, nothing is lost. However, careful analysis and data shows this to be a half-truth, at best.

The basic kinetic energy equation **$K.E. = 1/2mv^2$** indicates that the lost energy increases with the square of the speed.

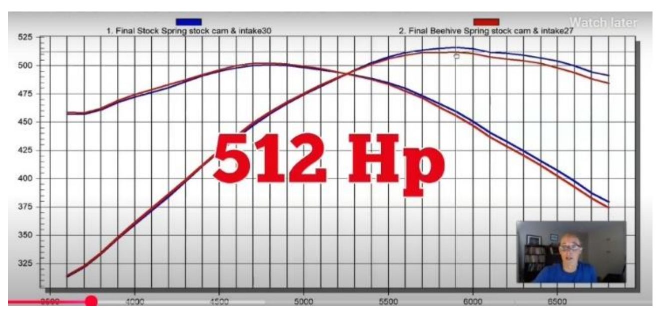
At some point therefore, the flywheel effect, no matter the magnitude, simply cannot cancel out the localized inertial spikes causing excess bearing loading & component deflection leading to localized heating due to hysteresis.

As engine speeds & power go up the proportional amount of energy lost to generic friction goes down whereas the reciprocating mass cost is inverted and goes up. For a typical reciprocating piston engine with a 4” stroke, the reciprocating mass energy cost is virtually nil below 3000 rpm but would exponentially increase to the square of the speed thereafter. Since most efficiency analysis focuses on conditions below 3000 rpm the reciprocating mass energy factor is of little concern as a practical matter. But when measuring for peak hp/speed against zero (0) reciprocating mass it is a real & consequential number necessarily factored for comparative accuracy.

Quantifying that number is not without challenges, however.

Ideally, we would quantify that number directly by removing the head, piston rings & oil pan (for windage) and accelerating and holding a reciprocating assembly @ full max hp speed. Then all the reciprocating mass is shifted to unidirectional constant speed (fully rotational) kinetic mass equalizing the total, accelerate and hold @ max hp speed. Then reduce the total kinetic mass by the amount normally needed to control all the removed reciprocating mass, accelerate and hold @ max hp speed. The difference in the observed load for each would be the clear and accurate isolated reciprocating energy cost both in transient and various steady state conditions including maximum hp/speed. Not being aware of any such test, the next best thing is to is to extrapolate a number from actual tests conducted that isolate the energy cost to control the reciprocating mass of a poppet valve train.

While waste heat generated by valve springs dedicated solely to controlling reciprocating mass @ ½ speed of the engine has been observed and documented many times it would appear that no one had isolated this specific energy loss in a full temp full load running engine until **Richard Holdener** did it in June of 2021 (See YouTube). All the parameters of the test were objective and favor accuracy. Mr. Holdener had nothing to gain by showing that a major aftermarket engine product actually lost power. The direction of the increased valve spring pressure & rate in the 2nd test would only increase frequency and reduce any bad harmonics. It appears clear that there were no such issues with the baseline test as well as the fact it was based on a proven and well-developed OEM setup.



[7]Figs. 5 & 6 Credit: Richard Holdener – June 2021

Once we know how much energy is lost due to a certain value of reciprocating spring load, we can calculate the amount of energy that is absolutely required to control just the reciprocating mass motion of the reciprocating portion of the valvetrain @ max hp ½ speed in the first instance. In this case contact friction is discounted because not only is it very low but what little there is may actually reduce the load on the springs.

The observed number is approximately a minimum of **7 hp** @ 5900 rpm for the 6.2L LS3 with a max mass production HP value of approximately 485 which equals a **1.44%** lost energy cost. Even with a 25% error factor (to compensate for any difference the crank-con-rod motion) this = a minimum of over **1%.**

Knowing that pistons and the upper portion of the conrods (above the CG) represent at least 50% more mass than all the valve train components/cyl in the same engine the 1% minimum is multiplied by 1.5. That 50% higher mass also reciprocates at twice the speed of the valve train. Since the inertia effects increase to the square of the speed that **= $1.5 \times 2^2 = 6 + 1_{vt}$ = 7%** minimum reciprocating mass energy cost. This = a reciprocating mass factor of **.93.**

Some would argue that if this were true then we should see the BTE drop after 3000 rpm by as much as 20% @ the max HP speed (See Fig. 15). However, this is not necessarily the case for 2 main reasons:

1. The portion of friction loss drops as speed/hp increases [4], and,
2. increased cycle speed reduces heat rejection as discussed herein.

Therefore, the base BTE of 35% could theoretically increase to **42%** (@ max speed/load) just by eliminating all the reciprocating mass.

6. [1]Friction & Incremental Losses:

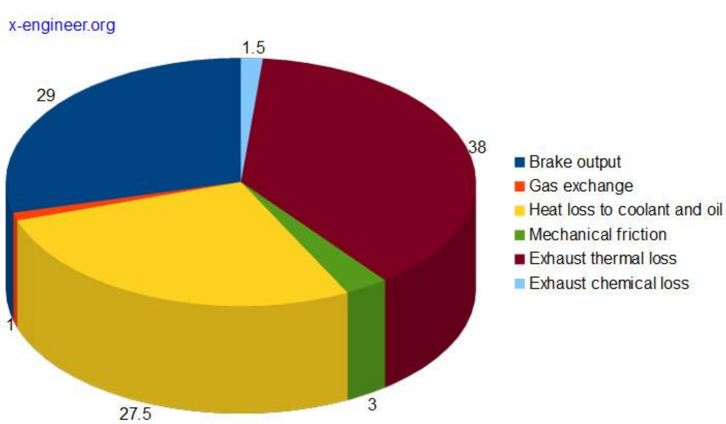


Fig. 7 Image: Engine losses at full load[4]
Credit: FEV

For the typical 4-stroke ICE with a base peak BTE of 29-35% the total mechanical friction loss @ full load and speed is approximately 3% (See Fig 7). This includes piston rings, piston to cylinder side loading, bearings the valve-train (=direct contact friction), Etc. It also includes auxiliaries such as the alternator (needed run the fuel pump, FI and spark), oil pump & water pump. For direct injection a high-pressure mechanical fuel pump is added that increases that number to approximately 4% or higher.

The TTX eliminates 80+% of all direct contact friction, has no water pump nor an oil pump (& eliminates 90% of the PT pumping losses). It will however increase the electrical demand to run the fuel pump, FIs and spark by 4-5X. This could result in an 80 amp draw at peak load & speed. However, that is equivalent to only 1.5 HP for a combined frictional load of less than 3 HP or an equivalent of .75%. This directly translates to an approximate 7% increase in the BTE @ PT. This also leaves enough energy to power light intake boost @ higher loads and still stay well within the 3% range relative to a 485 HP 4-stroke piston ICE. However, that number must be corrected to the projected power density this model predicts which nearly evaporates the entire friction load to .0015% or less @ 12,000 rpm (.003 @ 6,000 rpm).

This means that while the piston ICE has a final max power density that must be multiplied by a factor of .97 to account for mechanical friction losses and the TTX must be multiplied by a factor of .997 or a difference of 2.7% @ match speed of 6000 rpm (or .9985 @ double TTX rpm speed).

Being a direct swap from thermal waste to work increase means that we add that number directly to the BTE. So, the **42%** BTE now becomes **44.7%** BTE @ match speed & **44.85** BTE @ double speed/power.

7. **Gas Exchange & Chemical Loss:**

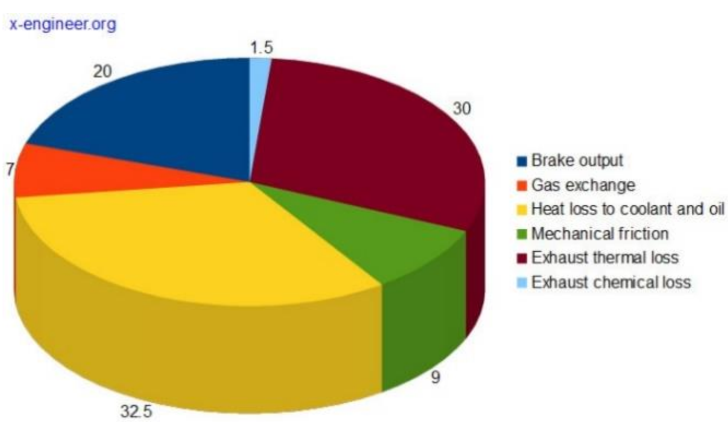


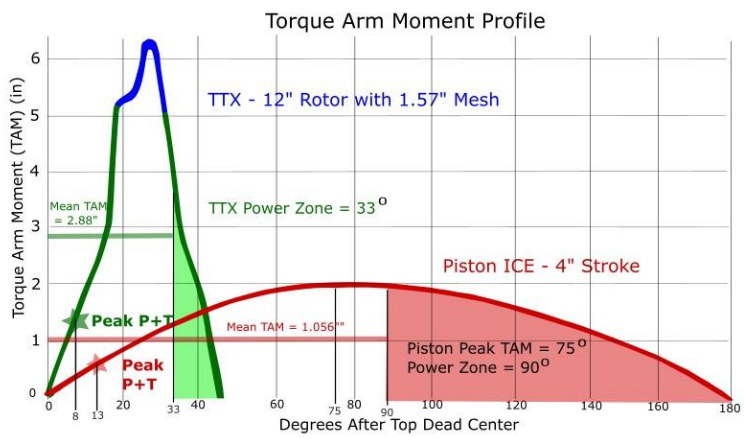
Fig.8 Image Credit: FEV
Engine losses at part load[4]

There is another 2.5% @ 100% load piston to be gleamed from eliminating the gas exchange loss and the exhaust chemical loss (See Fig. 8) (8.5% @ PT: See Fig. 6). Most of that and an additional portion of exhaust heat is recaptured by way of the **TTX turbo-flux and afterburner** cycles (Also can eliminate up to 8% gas exchange & chemical loss @ PT.- See Fig. 8) which for simplicity's sake is not part of the discussion here. Obviously, the proper function of these residual auxiliary cycles would have the effect of increasing the measurable power density and BTE beyond the numbers projected herein.

Cycle Dynamics, Torque Arm Moment (TAM) Profile & Expansion Work Conversion Efficiency:

In order to properly assess the true potential for each engine design, the model must define:

1. The difference in the length of time each PZ expansion cycle exerts an identical average *lb* force upon a crank/shaft (which favors the piston ICE):



As illustrated in Fig 9 the PZ cycle for the 4-stroke piston is 90° and the TTX is 33°. Therefore, an identical average *lb* force acting upon the average TAM for the entire length of the expansion cycle would result in a ft/lb/2π mean advantage for the piston of **2.72:1.**

Fig. 9

2. The length of the respective average TAM: As illustrated in Figs 4 & 9, the mean average TAM across the power zone for the 4-stroke piston with the 4" stroke is 1.056" (21mm). For the TTX it is 2.88" (71mm) which equals an advantage for the TTX of **2.72:1** which effectively cancels out No. 1 above**.**

3. The difference in rejectable heat based on the speed of each PZ cycle and its effect on both the average *lb* force applied to the crank/shaft and its ultimate effect on BTE:

The case for cycle speed v heat rejection:

Where there is a difference in the cycle speed and/or engine speed between comparative subjects the model must correct accordingly. Simple thermal dynamic principles dictate that any identical volume combustion chamber that is able to convert a given unit of fuel into a given quantity of expansion work and convert that expansion work into as much useful work as possible but do it in half the time will reject less heat to the chamber & active member(s) surfaces directly increasing the power density and BTE.

We measure heat current accordingly where:

1 Watt = 1 Joule/s

Therefore, we can measure the rate of energy (heat) transfer with the formula:

$$E_{(J)} = P_{(W)} \times t_{(s)}$$

Where **E** is the energy in Joules, **P** is the power in Watts, and **t** is the time in seconds (the time element being fundamental to these measurements).

The only other significant variable that comes into play is temperature differential (ΔT). Prioritizing power density the increased BTE will be slightly mitigated against the average temperature of the expansion gases across the PZ of the cycle which would necessarily increase the ΔT between them and the combustion chamber. However, prioritizing fuel efficiency the unit of fuel is reduced relative a given amount of useful work produced and average expansion gas temperature can remain constant or even be reduced reducing the ΔT and all emissions, including NO_X emissions, at the same time. Perhaps the best real-world case for quantifying this phenomenon in a running engine is the comparison of

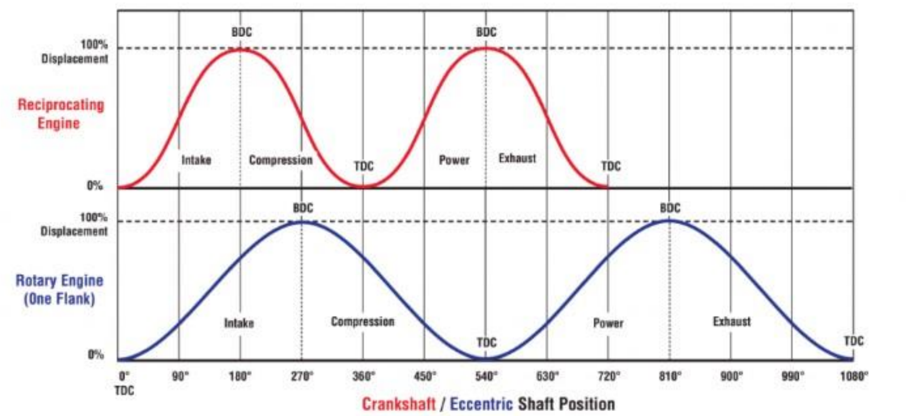


Fig. 10: Hard Media (Rotary v Piston-Aug 1, 2019)

our base generic subject, the 4-stroke piston ICE to the Wankel. Both of these subjects are mass produced (or producible) prime mover engine categories for which much research data exists. Among many differences that effect both power density and BTE is the 50% difference in cycle speed where the 4-stroke piston having an expansion cycle of 180 CA° and the Wankel 270 CA° respectively. Or put another more conservative way, the 4-stroke piston cycles 33.3% faster than the Wankel.

Few research papers isolate this Δ cycle speed issue and instead focus on other combustion related issues inherent to the Wankel such as CR, combustion chamber shape, burn rate, combustion efficiency, Etc. all of which do present some well documented challenges[3]. It is believed that the focus is on the combustion quality issues because that is the area for which much improvement is possible and has been shown whereas the cycle timing issue is largely unfixable. However, world renowned Wankel expert the former **Jim Mederer (1942-2016)** did state the following:

"The rotary also has 1.5 times as many milliseconds to transfer heat from the burning mixture into the oil and water. This is one reason why rotaries waste more heat in the process of staying cool."

Because of the large amount of data on the Wankel in comparison with the 4-stroke piston there is a verifiable test case to develop a **Cycle Speed Energy Conversion** (CSEC) model which can then be applied to engine designs where cycle speeds can be identified but for which no hard data yet exists. This would include the TTX. Where the initial working BTE for the Wankel is generously assumed to be 28%, the formula to quantify the change in BTE due to cycle speed alone is:

The corrected CSEC BTE = the slowest cycle - the fastest cycle ÷ the slowest X the initial portion of the total expansion work that is not converted to work + 1 X the base net BTE =

CSEC = (270° – 180°)/270° = .333 x .72 (-%BTE) = .24 + 1 = 1.24 x 28%η_{th} = 34.7% η_{th} (BTE)

Where the CSEC advantage factor is 1.24 which can be used to calculate BVPD later.

Correcting for the lower knock limit CR caused in large part by the increased (slow cycle speed) heat rejection (Approx. 9.7 vs 11:1 = - 1.8% BTE) a 6.7 basis point change in the BTE increases to 8.5. Correcting for the slightly higher friction loss of the piston 4-stroke due to the valve train and high piston to cylinder side loading the 8.5 number is reduced to approximately 7 BTE basis points[1]. This means that even if all the combustion issues could be solved completely the Wankel (& Liquid Piston) would still be approximately 25% less efficient @ the same rpm speed than a comparable 4-stroke piston engine attributable to the cycle speed difference alone despite the piston engine's higher internal frictional losses. Data confirms that the Wankel rotary is well behind the piston especially at slower speeds where the negative effects of the slower cycle speed become more pronounced.

The now largely verified CSEC formula adjusts to different cycle speed differentials and base net BTE by adjusting for changes in the initial BTE to -BTE ratio. Where the base net TTX BTE is **44.7%** =

CSEC = (90° – 33°)/90° = .633 x .553 (-%BTE) = .350 + 1 = 1.350 x 44.7% η_{th} = 60.3% η_{th} (BTE) @ the same rpm engine speed (CSEC factor = 1.35)

&

CSEC = (90° – 33°)/90° = .633 x .551 (-%BTE) = .349 + 1 = 1.349 x 44.85%η_{th} = 60.5% η_{th} (BTE) @ double rpm engine speed (CSEC factor = 1.349)

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[1] Both engines have reciprocating mass. The Wankel reciprocates elliptically which is why it is in the category of a "rotary" and not a "turbine". However, the elliptical form of reciprocation may somewhat mitigate the reciprocating energy cost by some unknown specific amount.

Speed over Speed:

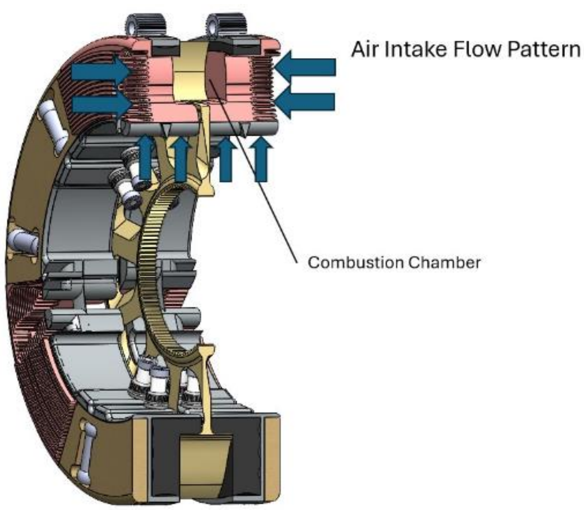
Obviously actual engine speed or rpm also has a direct effect on cycle speed. Therefore, where comparing engines with different attainable and reliable peak torque or hp speed this would also be a factor[2].

For example, the Wankel's slow cycle speed has the advantage of increasing the air flow capacity which should allow it to attain higher peak torque &/or hp speed. At 8000 peak torque rpm compared to the 6000 peak torque rpm for our 4" stroke 4-stroke piston the Wankel could close the BTE gap by 35% or bring it to 4.5 basis points vs 7 or 16% vs 25% reduced efficiency.

This phenomenon has been quantified and documented ala Stanten[5] for one example[5]:

> **"The slower engine speed increased both the convective and crevice heat loss due to the greater time available for heat transfer. The increase in the convective heat loss at the slower engine speed was roughly 14%"**

For another example, despite having major aspects of their design that is not focused on efficiency such as valve timing, *F1* engines can achieve as high as 52% BTE far higher than the highest efficiency focused mass production SI piston engines. The difference cannot be fully quantified to be from advanced fuel delivery and ignition systems alone, especially since many production engines have similar technology. Therefore, the ability to comfortably live in a rev range between 11 & 15,000 RPM is a factor and needs to be considered.



The TTX, with zero (0) reciprocating mass (& 3X air flow capacity, See Fig. 11 above) will have no issue running continuously in that rev range and higher with the result being even higher power density & BTE @ higher speeds.

Considering that the TTX is expected to be able to achieve at least 3X the peak torque-hp speed a 2X minimum speed analysis is in order. We can quantify the change in cycle speed due to engine speed (rpm) by simply cutting the TTX cycle in half or from 33° to 16.5°. Doing so gives:

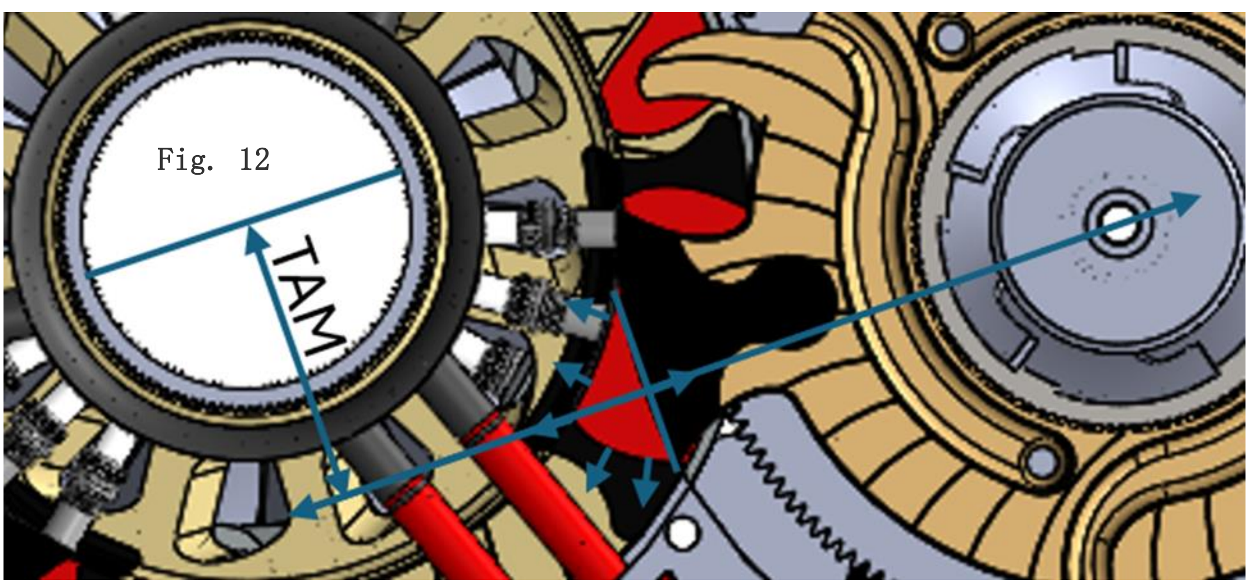
CSEC = (90° – 16.5°)/90° = .816 x .551 (-%BTE) = .450 + 1 = <u>1.450</u> x 44.85%η_{th} = <u>65</u>% η_{th} (BTE) @ double rpm speed (CSEC factor = <u>1.45</u>)

However, this quantified phenomenon is dependent on combustion speed or burn rate being able to keep up with an extreme level of cycle speed which may not be the case. Data developed by Honda in the late 60s & early 70s on a single sparkplug (SP) gas SI piston ICE suggests this is not likely to be a limiting factor for the TTX with four (4) SPs[6]. This is an additional reason why H_2 may be the ideal fuel for the TTX with 3X the burn rate of common liquid fuels.

4. The difference in the respective TAM profiles Including Effective TAM:

In piston ICE research, changes in both crank radius and rod length can be made to affect minor changes in the TAM profile most consequently by increasing or decreasing the piston dwell time around TDC. Research conducted by Honda in 2006 indicates that despite the change being minor, increasing the piston dwell time around TDC increased the heat loss because of the increased high ΔT time exposure[8] which further supports the cycle time thermal analysis above. Such a condition normally increases the potential for detonation (knock). Therefore, the inverse is established and would directly increase the knock limited CR. The optimum TAM profile therefore would increase volume and the TAM leverage very rapidly after TDC but also increase dwell time in the high TAM region.

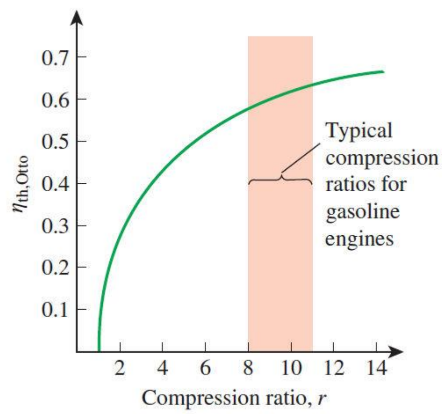
Observing the TAM profile comparison in Figs. 4, 9 & 12 it is clear that the TTX represents a fairly profound improvement in the TAM profile over the typical 4(& 2)-stroke piston ICE. Based on this and the known data, the TTX knock limited CR can be increased from 11:1 to a minimum of 15:1. Comparing each engine type to its own knock limited CR is not just for the sake of accuracy alone as the inherent high speed cycling of the TTX must adjust its CR in part to maintain proper combustion efficiency and burn rate.



Accordingly, to calculate the BTE & BVPD change with CR we use the formula:

$$\eta_{th\ \&/or\ BVPD} = 1 - \left(1/R^{(k-1)}\right)$$

Using the coefficient of adiabatic expansion of air @ 1.35 for k[2] where R = the CR, an increase from 11:1 to 15:1 yields a power and BTE increase of **4.5%** (See Fig. 13).



It should be noted that some advocate for a different CR formula to isolate the BTE based on CR change. Fig. 14 below is a chart based on that formula. It shows a 60% difference in the BTE factor from 11:1 to 15:1 CR @ **7.2%** vs the power factor for the same change @ **4.5%** (also verified by Hard Media). However,

THERMAL EFFICIENCY FACTOR

NEW COMPRESSION RATIO

ORIG C.R.	7.5:1	8.0:1	8.5:1	9.0:1	9.5:1	10.0:1	10.5:1	11.0:1	11.5:1	12.0:1	12.5:1	13.0:1	13.5:1	14.0:1	14.5:1	15.0:1
7.5:1	100.0%	102.1%	103.9%	105.7%	107.3%	108.8%	110.2%	111.5%	112.7%	113.8%	114.9%	116.0%	116.9%	117.8%	118.7%	119.6%
8.0:1	98.0%	100.0%	101.9%	103.6%	105.1%	106.6%	107.9%	109.2%	110.4%	111.5%	112.6%	113.6%	114.6%	115.5%	116.3%	117.1%
8.5:1	96.2%	98.2%	100.0%	101.7%	103.2%	104.6%	106.0%	107.2%	108.4%	109.5%	110.6%	111.5%	112.5%	113.4%	114.2%	115.0%
9.0:1	94.6%	96.6%	98.4%	100.0%	101.5%	102.9%	104.2%	105.5%	106.6%	107.7%	108.7%	109.7%	110.6%	111.5%	112.3%	113.1%
9.5:1	93.2%	95.1%	96.9%	98.5%	100.0%	101.4%	102.7%	103.9%	105.0%	106.1%	107.1%	108.1%	109.0%	109.8%	110.7%	111.4%
10.0:1	91.9%	93.8%	95.6%	97.2%	98.6%	100.0%	101.3%	102.5%	103.6%	104.7%	105.6%	106.6%	107.5%	108.3%	109.1%	109.9%
10.5:1	90.8%	92.6%	94.3%	95.9%	97.4%	98.7%	100.0%	101.2%	102.3%	103.3%	104.3%	105.2%	106.1%	107.0%	107.8%	108.5%
11.0:1	89.7%	91.6%	93.2%	94.8%	96.2%	97.6%	98.8%	100.0%	101.1%	102.1%	103.1%	104.0%	104.9%	105.7%	106.5%	107.2%
11.5:1	88.7%	90.6%	92.2%	93.8%	95.2%	96.5%	97.8%	98.9%	100.0%	101.0%	102.0%	102.9%	103.8%	104.6%	105.4%	106.1%
12.0:1	87.8%	89.7%	91.3%	92.8%	94.2%	95.6%	96.8%	97.9%	99.0%	100.0%	101.0%	101.9%	102.7%	103.5%	104.3%	105.0%
12.5:1	87.0%	88.8%	90.4%	92.0%	93.4%	94.7%	95.9%	97.0%	98.1%	99.1%	100.0%	100.9%	101.7%	102.5%	103.3%	104.0%
13.0:1	86.2%	88.0%	89.6%	91.1%	92.5%	93.8%	95.0%	96.1%	97.2%	98.2%	99.1%	100.0%	100.8%	101.6%	102.4%	103.1%
13.5:1	85.5%	87.3%	88.9%	90.4%	91.8%	93.0%	94.2%	95.3%	96.4%	97.4%	98.3%	99.2%	100.0%	100.8%	101.5%	102.3%
14.0:1	84.9%	86.6%	88.2%	89.7%	91.0%	92.3%	93.5%	94.6%	95.6%	96.6%	97.5%	98.4%	99.2%	100.0%	100.7%	101.5%
14.5:1	84.2%	86.0%	87.6%	89.0%	90.4%	91.6%	92.8%	93.9%	94.9%	95.9%	96.8%	97.7%	98.5%	99.3%	100.0%	100.7%
15.0:1	83.6%	85.4%	86.9%	88.4%	89.7%	91.0%	92.2%	93.2%	94.3%	95.2%	96.1%	97.0%	97.8%	98.6%	99.3%	100.0%

Fig. 14 – Credit: Hard Media

many just use the same more conservative formula for both, which, is the method chosen for this model so as to error on the side of moderate & most credible change.

In such a case the match engine rpm speed = 60.3% x 1.045 = **63%** BTE.

In the double engine rpm speed case = 65% x 1.045 = **68%** BTE

[2] This assumes that the mean PZ cycle combustion gas T would be identical between the Piston CR @ 11:1 and the TTX CR @ 15:1 for a given mean force when in fact the TTX will be lower based on compression cycle efficiency & speed, PZ expansion cycle efficiency & speed, lower heat rejection & lower T spikes due to an improved TAM profile. Adjusting the coefficient of adiabatic expansion of air to account for the specific mean combustion T for each case could increase both BTE & power density for the TTX an additional 1% @ PT.

As illustrated in Figs. 4 & 9, the most inefficient part of the typical 4-stroke piston TAM profile is in the first 25° CA° ATDC. This is where the highest **ΔT** matches with very low TAM leverage resulting in the highest heat rejection of the cycle. Consequently, any improvement in the initiating TAM profile has the direct effect of increasing both power density and BTE beyond and separate from the average TAM alone. The goal is to decrease the peak chamber pressure relative to a fixed average (or the inverse). In this way expansion work is shifted away from low TAM high heat rejection zone to the higher TAM lower heat rejection zone. This results in a higher average TAM relative a given pressure or a higher TAM to pressure ratio. In this manner every **1%** drop in peak pressure relative to a fixed average pressure throughout the PZ results in a **2%** positive change in BMEP and BTE. It has the effect of increasing the average TAM relative average pressure beyond what the change in active power member geometry, including CR, would indicate in isolation.

This also means less chance for harmful knock, less noise, less overall emissions and in particular less NO_x.

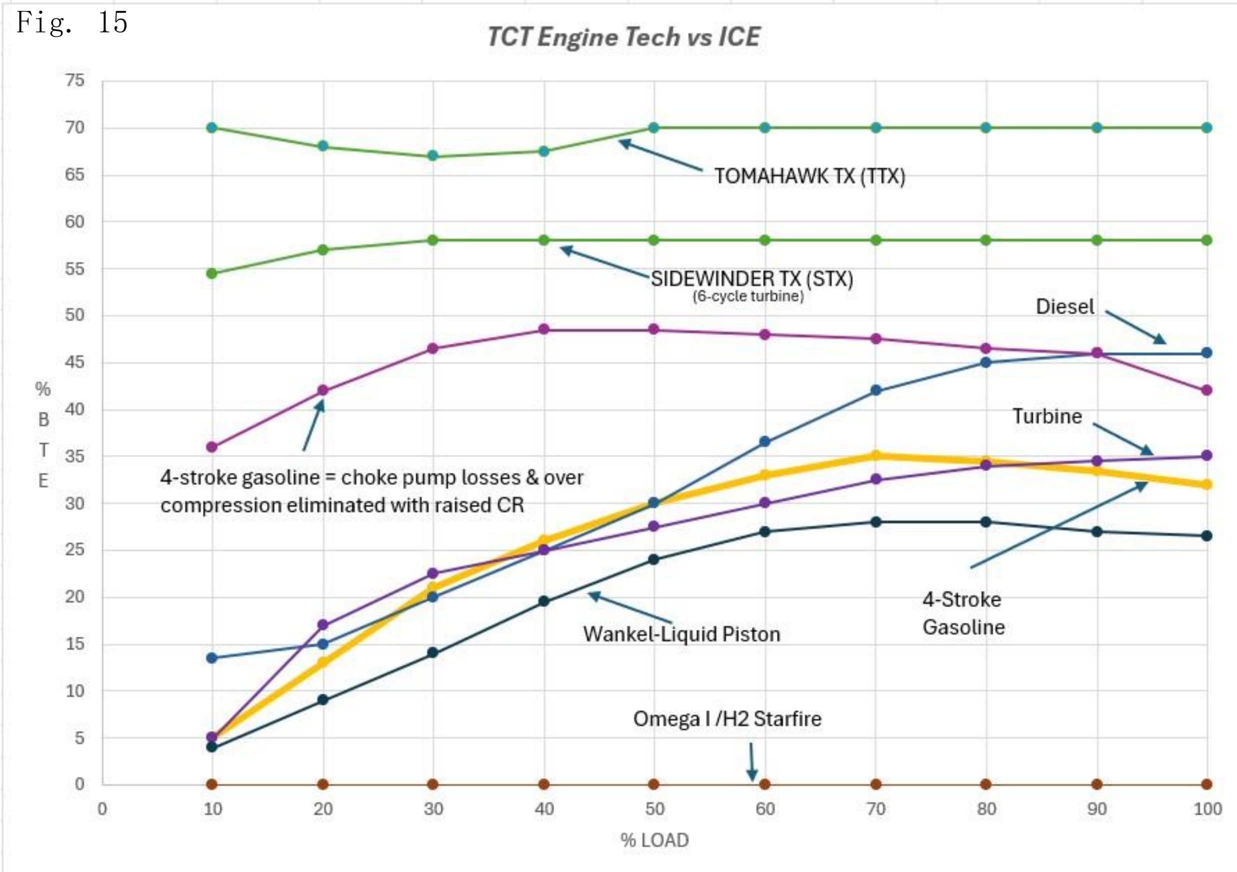
As is clear from the diagram in Fig. 4 & 8 the difference in the TAM:P ratio is significant and based on real measurements (See Fig. 12). But, until an actual prototype is built, tested and tuned the actual operational change can only be estimated. The TTX TAM @ peak pressure & temperature is at least 2.5X greater than the 4-stroke piston. Unless the peak pressure is lowered by the same amount this will result in a substantial increase in more useful work being produced and more efficiently, independent of increased CR and other factors. Based on this a TAM:P ratio is increased by a minimum of 3% resulting in an overall TAM profile efficiency enhancement minimum of 6% fully independent of the overall cycle speed efficiency difference.

The final TTX BTE result can be projected to be approximately:

In the match engine rpm speed case = 63% x 1.06 = **66.8%** BTE, AND

in the double engine rpm speed case = 68% x 1.06 = **72%** BTE.

Fig. 15



Keeping in mind that this does not include the TTX Turbo-Flux and/or afterburner cycles which will increase both BTE & BVPD.

Brake Volumetric Power Density - Single Thermodynamic Cycle:

We can use much of the model work developed for defining true BTE potential to calculate the ultimate BVPD for both of our test subjects.

Elements:

The isolated BVPD potential formula for a single thermodynamic expansion cycle for each engine is defined as:

4" Stroke Piston ICE BVPD = $P/V_{Up} = (\max T_{\omega}/5252_{\omega}) \cdot \eta_{V} \cdot (W_{I} = 1) - W_{Ff}) \cdot (W_{I} = 1) - E_{\sim\omega 2})$

AND

TTX BVPD = $P/V_{Up} = (\max T_{\omega}/5252_{\omega}) \cdot \eta_{V} \cdot (W_{I} = 1) - W_{Ff}) \cdot (W_{I} = 1) - E_{\sim\omega 2}) \cdot (W_{I} = 1) + (\eta_{CR}) \cdot$

$(W_{I} = 1) + (TTX_{TAM})_{\eta}) \cdot (W_{I} = 1) + \text{TTX } e_{\omega\eta})$

Where:

P = Power (pressure)

V = Volume (reference volume of single (series) compression & single combustion chamber)

η_V = Volume (volumetric) efficiency

η_{CR} = Compression Ratio efficiency

W = Work (Watt) & W_I = Indicated Work (for 4” stroke 4-stroke piston ICE) & W_F = Friction Work

K_m = Kinetic Mass

E = Energy

F = Force

F_f = Friction Force

$_m$ = Mass

***e* = Expansion (expansion event/cycle)**

T = Torque (or temperature)

t = Time

ω = Speed (rpm)

$_\eta$ = Efficiency (%)

2π = 1 x 360°

$_{U\rho}$ = Potential Power Density Factor of 1

***TAM* = Torque Arm Moment**

$\tilde{\cdot}$ = Reciprocal Kinetic Mass Motion

The essential elements are:

Piston BVPD = P/$V_{U\rho}$ = (Engine Peak Torque Speed) · (Volumetric Efficiency) · (Piston ICE Friction Factor Of 1) · (Piston ICE Reciprocating Kinetic Mass Energy Loss Factor Of 1)

AND for the TTX add · **(Indicated Piston Work + TTX T.A.M. Profile Efficiency as a % over 1) · (Indicated Piston Work + TTX Expansion Cycle Speed Efficiency as a % over 1) · (Indicated Piston Work + TTX CR Efficiency as a % over 1)**

Therefore: **Piston BVPD = P/$V_{U\rho}$ = (6000 peak torque rpm/5252 = 1.14) x .95 x .97 x .93 = .976**

&

TTX BVPD = P/$V_{U\rho}$ = (6,000 peak torque rpm/5252 = 1.14) x .98 x .997 x 1 x 1.06 x 1.35 x 1.045 = 1.66

Therefore, a single TTX expansion cycle/**2π** with the exact same functional parameters is equal to 1.70 Piston expansion cycles/**2π** measured @ the same engine rpm speed.

Double engine rpm speed = TTX BVPD = P/$V_{U\rho}$ = (12,000 peak torque rpm/5252 = 2.284) x .99 x .9985 x 1 x 1.06 x 1.45 x 1.045 = 3.62/2 = 1.81

Therefore, a single TTX expansion cycle/2π with the exact same functional parameters is equal to <u>1.85</u> Piston expansion cycles/2π measured @ double engine rpm speed. This reflects a slight decrease in the proportional friction loss & blowby to volumetric efficiency as well as the speed over speed cycle efficiency advantage.

Cycle Propagation through Frequency & Speed:

We use:

$$\mathbf{BVPD = P/V_{U\rho} = (max\ T_{\omega}/5252_{\omega}) \cdot V_{U\eta} \cdot (W_I(=1) - W_{Ff}) \cdot (W_I(=1) - E\dot{\sim}_{\omega 2}) \cdot (W_I\,(TAM)_{\eta}) \cdot (W_I\, e_{\omega\eta}) \cdot e_f}$$

To calculate the difference based on the frequency of completed expansion cycles per 2π or a single 360° revolution where e_f = Expansion Cycle Frequency we take the calculated piston BVPD factor of .976 x .5 which = .488, and we take the calculated match rpm engine speed TTX BVPD factor of 1.7 x 3 which = 5.1. This renders a BVPD ratio of <u>10.4:1.</u>

However, the calculated flow capacity indicates the TTX will be able to maintain volumetric efficiency @ 3X the peak torque engine rpm speed as the 4" stroke NA piston. Therefore, a minimum double speed comparison is justified and otherwise in order (Fig. 11).

The calculated double speed TTX BVPD factor of 3.62 x 3 which = 10.86. This renders a BVPD ratio of <u>22.2:1.</u>

We can assume different volumetric efficiencies based on different levels of leakage or blowby by simply adjusting the η_v value. Therefore, if we assume the blowby is 10X the ringed piston the η_v value goes to .91 (because 1% blowby is already counted @ 12,000 rpm) which = 9.89 or over 20X the piston.

Conclusions:

Predictive models of variations of standard well developed types of PD ICEs have proven to be useful tools in the development of the ICE for many decades.

These models, however, are both incomplete & inadequate to reliably or accurately predict new limits exposed by a radical departure from mere variations in standard types of ICEs such as that presented by the PD centra-fusion multi-turbine or Tomahawk TX (TTX).

Therefore, a specifically tailored predictive model has been developed to allow for an accurate, within reasonable margins of error, mathematical prediction of both the brake thermal dynamic efficiencies (BTE) & Brake Volumetric Power Density (BVPD) of the TTX compared to the general broad type of ICE known as the Otto Cycle 4-stroke Piston ICE.

The new model indicates that substantial gains in BTE are possible due in large part to the complete elimination of reciprocating mass, improvements in the TAM profile and both CA° based expansion cycle speed & attainable & maintainable peak torque/hp engine rpm speed. Based on this the model indicates that the SI Otto 4-stroke reciprocating piston single cycle BTE can be improved from 35% to a range of **66.8 - 72%** depending on speed before considering the turbo-flux & afterburner cycles which is a more effective form of compounding where it is integrated internally reducing heat lost to both surface area & time.

The model further indicates that the net work producible from a single expansion cycle/360° can be increased with the TTX design by **70 - 85%.**

However, the compounded effects of TTX cycle speed efficiency allow for the improved expansion cycle’s frequency to be increased by a magnitude 6X resulting in a **10.4:1** BVPD ratio @ the same engine rpm speed.

In addition, the **TTX’s** attainable and maintainable engine rpm speed (w/SI @ 4XSP/CC) can be increased by a magnitude 3X indicating a reliable magnitude 2X minimum resulting in an over **22:1** BVPD ratio.

The new model reflects a simple underlying formula where a mechanism is created to substantially increase the speed & efficiency of a single expansion cycle/360° and simultaneously fully exploits the resultant increased cycle speed creating a “hyper-cycle’ to propagate more hyper-cycles within a standard space in time (**2π**) unit without some form of internal gear reduction which would only serve to swap out engine rpm speed & HP for torque. Combined with higher engine rpm speed, seemingly impossible increases in both power & efficiency become feasible as the cognizable mathematical model predicts.

Lacking any substantive sealing issues, the challenge of maintaining the proper intermesh timing control between rotors under all conditions becomes much less daunting.

In both real & mathematical terms, the TTX represents a potential monumental leap in ICE technology that warrants aggressive & rapid development.

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