A

Seminar report

On

Ball Piston Engine

Submitted in partial fulfillment of the requirement for the award of degree
Of Mechanical

SUBMITTED TO: **SUBMITTED BY:**

www.studymafia.org www.studymafia.org

Preface

I have made this report file on the topic **Ball Piston Engine**; I have tried my best to elucidate all the relevant detail to the topic to be included in the report. While in the beginning I have tried to give a general view about this topic.

My efforts and wholehearted co-corporation of each and everyone has ended on a successful note. I express my sincere gratitude to …………..who assisting me throughout the preparation of this topic. I thank him for providing me the reinforcement, confidence and most importantly the track for the topic whenever I needed it.

Acknowledgement

I would like to thank respected Mr…….. and Mr. ……..for giving me such a wonderful opportunity to expand my knowledge for my own branch and giving me guidelines to present a seminar report. It helped me a lot to realize of what we study for.

Secondly, I would like to thank my parents who patiently helped me as i went through my work and helped to modify and eliminate some of the irrelevant or un-necessary stuffs.

Thirdly, I would like to thank my friends who helped me to make my work more organized and well-stacked till the end.

Next, I would thank Microsoft for developing such a wonderful tool like MS Word. It helped my work a lot to remain error-free.

Last but clearly not the least, I would thank The Almighty for giving me strength to complete my report on time.

CONTENT

ABSTRACT

A patented new power machine concept has been designed and analyzed for production, and proof of principle subscale tests have been performed, with positive results. The machine design concept is applicable as a compressor, pump, motor, or engine. Simplicity of design based on spherical ball pistons (Figures 1 and 2) enables a low moving part count, high power to weight ratio, elimination of valve train and water cooling systems, and perfect dynamic balance.

The new design concept utilizes novel kinematic design to completely eliminate inertial loads that would contribute to sliding friction. Also, low leakage is maintained without piston rings by using a small clearance on the ball piston, resulting in choked flow past the ball. These features provide the potential for an engine with higher efficiency than conventional piston engines. The engine design utilizes existing recent technology to advantage, such as silicon nitride ball pistons, so a large development effort is not required.

INTRODUCTION

Efforts to develop rotary internal combustion engines have been undertaken in the past, and are continuing. One main advantage to be gained with a rotary engine is reduction of inertial loads and better dynamic balance. The Wankel rotary engine [2] has been the most successful example to date, but sealing problems contributed to its decline. The Hanes rotary engine [3] uses an eccentric circular rotor in a circular chamber with sliding radial vanes. This engine has never been fully tested and commercialized, and has a sealing problem similar to that of the Wankel. A more recent development, the Rand Cam engine [4], uses axial vanes that slide against cam surfaces to vary chamber volume. Currently under development, it remains to be seen whether the Rand Cam can overcome the sealing problems that are again similar to those of the Wankel.

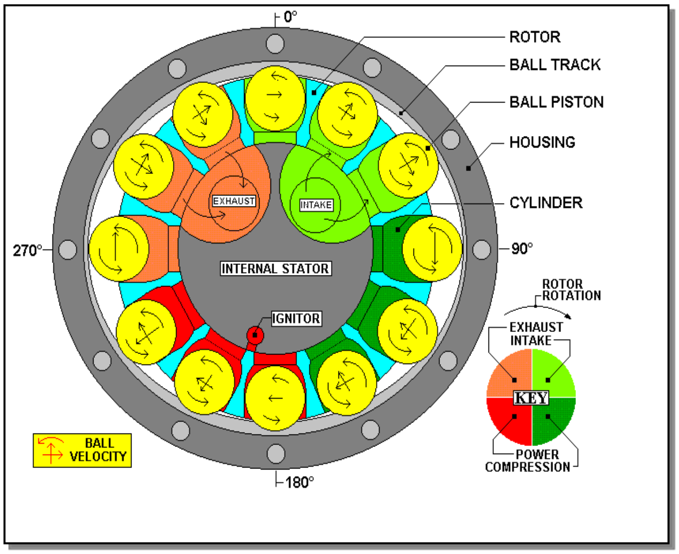

Figure 1. End section view of engine design

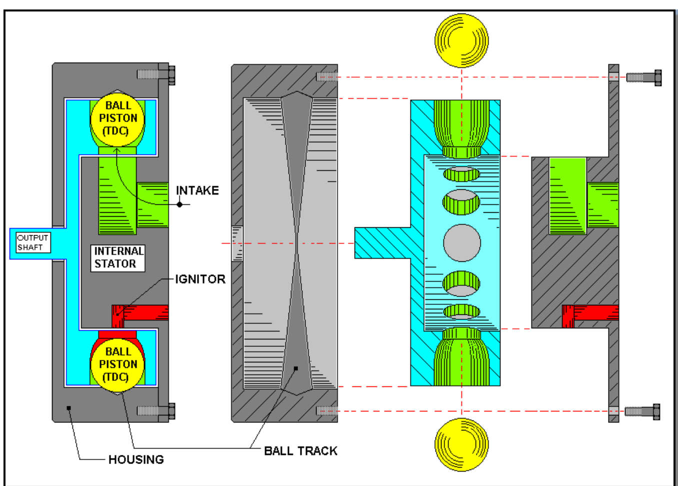

Figure 2. Side exploded section view of engine design

In the compressor and pump arena, reduction of reciprocating mass in positive displacement machines has always been an objective, and has been achieved most effectively by lobe, gear, sliding vane, liquid ring, and screw compressors and pumps [5], but at the cost of hardware complexity or higher losses. Lobe, gear, and screw machines have relatively complex rotating element shapes and friction losses. Sliding vane machines have sealing and friction issues. Liquid ring compressors have fluid turbulence losses.

The new design concept of the Ball Piston Engine uses a different approach that has many advantages, including low part count and simplicity of design, very low friction, low heat loss, high power to weight ratio, perfect dynamic balance, and cycle thermodynamic tailoring capability. These aspects will be discussed in more detail below.

THE DESIGN CONCEPT

Although the design is applicable as a compressor, pump, or engine, the engine implementation will be used for concept discussion. Figures 1 and 2 show end and side cross section views, respectively, of a four stroke engine design.

Mode of operation **-** The basis of the design is ball pistons rolling on an eccentric track. The balls exert tangential force on the cylinder walls which turn the rotor. Useful power is available at the rotor output shaft. The combustion chambers are within the spinning rotor. Chamber porting for intake, compression, power, and exhaust strokes is achieved by passage of the chamber tops across an internal stator with appropriate feeds as the rotor spins.

Beginning at top dead center (TDC) at 0 degrees rotation, the stator intake passage is open to the cylinder and a fuel/air charge is pulled into the cylinder as the ball piston moves radially outward for the first 90 degrees of rotation (intake stroke).

Then the intake passage is closed off, and the ball reverses radial direction for the next 90 degrees of rotation, during which time the new charge is compressed (compression stroke).

Just past 180 degrees rotation, the compressed charge is ignited as the cylinder port passes a small ignitor port. Combustion ensues, and the high combustion pressure pushes radially outward on the ball piston for the next 90 degrees of rotation. The ball in turn pushes tangentially on the cylinder wall because of the “slope” of the eccentric ball track, which is now allowing the ball to move radially outward. The tangential force produces useful torque on the rotor (power stroke).

At 270 degrees of rotation, the spent combustion charge is allowed to escape through the exhaust passage as the cylinder port is uncovered. Exhaust is expelled as the ball moves radially inward for the next 90 degrees of rotation (exhaust stroke). Then the cycle repeats.

Important Design Features - The basic operation of the new design is conventional for an internal combustion engine, i.e. a piston reciprocates within a cylinder, and with porting, implements the four strokes of the Otto cycle. However, there are a number of features that make this engine design favorable for high efficiency and emissions control.

The porting required for four stroke operation is achieved with no additional moving parts, and no valve train losses. The porting mechanism is achieved with simple port clocking within the rotor/internal stator bearing interface. Thus, part count is low and the hardware is simple in geometry, with only the rotor and ball pistons as moving parts.

Note that cylinder induction and mixing are aided by centrifugal and coriolis accelerations, because the cylinders are within the spinning rotor.

Sliding friction sites are minimized by the use of a rolling ball piston. Friction at conventional piston rings, piston pin, and connecting rod/crankshaft bearing are eliminated. Sliding friction still exists at the ball/cylinder wall contact, but is minimized by special material selection and working gas hydrodynamics (and possibly local lubrication). The rotor/stator bearing is of a gas or fluid hydrostatic type, so friction is very low at that site.

The use of an eccentric ball track allows tailoring of the chamber volume vs. time to optimize the cycle from a thermodynamic and chemical kinetics standpoint. The only requirement is that the ball return to the starting radius at TDC before intake. For example, the expansion/exhaust stroke length can be made different than for intake/compression for more exhaust energy recovery, or the combustion can be held at constant volume for a certain period.

Multi-cycle rotors can be implemented. Instead of 4 strokes, 8, 12 or more strokes can be traversed in a single revolution. Compressors and pumps can use any multiple of 2 strokes (intake and compression only), either in parallel or staged arrangement. Provided that inertial forces are controlled (to be discussed later), power to weight ratio can therefore be made high.

Other engine configuration options are also under investigation, including a dual rotor/intercooler configuration, diesel cycles, and 2 stroke cycles. The dual rotor option is attractive because it allows the compression and expansion ratios to be widely different (on separate rotors), but there are pumping losses that must be considered.

The use of many ball pistons, which each undergo the four strokes in clocked fashion, results in smooth power delivery and small net oscillatory forces. In fact, the total ball inertia forces are automatically balanced by symmetry if the number of balls is even. Further, combustion forces can be balanced by using an eight stroke rotor or stacking rotors axially with realtive clocking. Also note that a four (or higher) stroke rotor compressor would be balanced.

Novel design of the ball track has been devised that will eliminate inertial forces on each ball that contribute to friction. As the ball moves in and out radially on the eccentric track while the rotor spins, coriolis and other acceleration forces are generated on the ball radially and tangentially. Net tangential inertial forces contribute to friction at the ball/cylinder wall contact point. By changing the ball rolling radius using a widening/narrowing dual contact track in a prescribed manner, Figure 3, the net tangential inertial forces on the ball can be eliminated. In essence, the track design results in a balance of translational and rotational ball kinetic energy to eliminate tangential force. In other words, the ball track is designed so that the ball rolls around the track in synchronization with the rotor at constant rotation rate. Due to the form of the laws of motion, it is possible to maintain this condition at all rotation rates with a fixed track design. This allows the machine to be run at any high rpm desired, until the mechanical limits of the ball piston rolling on the track are reached (Hertzian stress fatigue). Engine power theoretically increases linearly with rpm. In actuality, intake flow dynamics may limit peak power at very high rpm, but that depends on the intake passageway details.

There is another interesting by-product of the rolling ball approach. The ball spins at very rates around its own axis, while it is radially compressed by centrifugal forces of rotation about the rotor axis. These two sources of inertial load tend to cancel out in terms of generating internal ball stresses. This allows high engine speeds to be sustained with less ball fatigue damage.

Heat loss is kept low because the engine intake can be configured to flow through the outer stator/rotor cavity. Rotor heat loss is gained by the intake charge, with less loss to the outer stator.

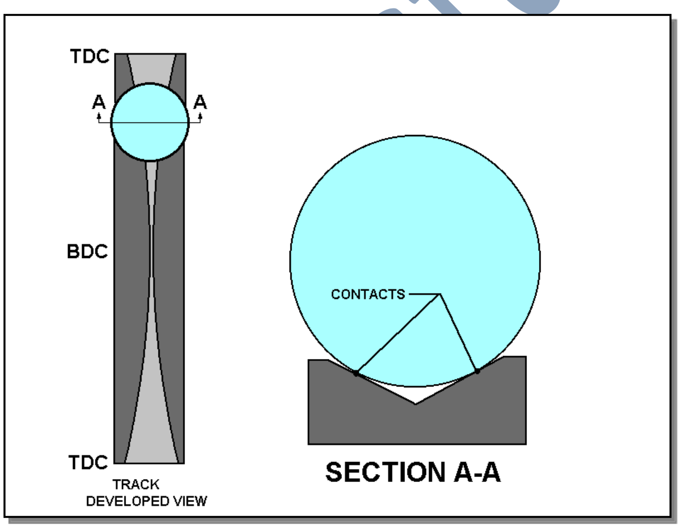

Figure 3. Dual contact variable rolling radius ball track concept

Technical Challenges - The main concerns for operation of the new machine are being addressed in focused subscale testing.

First, leakage through the ball piston/cylinder gap is a significant factor for engine or compressor efficiency, especially at low speeds. Calculations show that the flow is choked during combustion due to high pressure differential and small clearance area. Choking is helpful in keeping leakage to acceptable levels. Engine efficiency predictions based on simple choked flow leakage models are very favorable. Leakage tests performed in subscale testing have shown that leakage is less than the simple models predict, and dependence on ball spin, pressure, and rpm have been and are being characterized.

Second, the friction and wear at the ball piston/cylinder wall sliding interface is important. Engine performance depends on the magnitude of the effective friction coefficient, and high relative sliding speed can contribute to wear. Engine efficiency predictions based on an average friction coefficient of 0.1 or less are very favorable. Subscale tests have proven that the coefficient of friction for a silicon nitride ball piston on polished steel with no lubrication is about 0.075 ± 0.03, about the same as estimated.

The wear issue must be proven out mainly by testing with a full range of operating conditions. Thus far, tendency for cylinder wall plasticity has indicated that cylinder material must be of high hot strength and hardness. Large reductions in “wear-in” plastic flow were achieved by changing cylinder walls from 1018 hot rolled steel to 17-4PH hardened to about R_c 44. A material with better hot hardness, such as achievable with M2 high speed tool steel, has been subsequently selected to resist high sliding flash temperatures and completely eliminate cylinder wall plastic deformation. Low cost production options include case hardening, plating over a hot hard substrate, coatings, and other surface treatment technologies.

It is intended to design the machine for no lubrication, except that available from the working gas or fluid. This is most feasible for compressor and pump applications. However, lack of lubrication is a driving consideration in cylinder wall material selection for the engine, based on subscale testing to date with air only. Extra lubrication is a secondary design option that may be best for some applications, especially the engine, where loads are higher. Lubrication can reduce friction coefficient and wear potential and provide hydrodynamic separation at the ball piston/cylinder wall, and also can reduce leakage flow past the ball piston. However, there will be a trade off for residue build up, emissions, and maintenance.

INERTIAL CONTROL THEORY

Early efforts to analytically demonstrate engine performance were plagued by excessive frictional losses due to large coriolis forces on the ball. Although the effect was conservative, i.e. average tangential force per revolution of the rotor was zero, the attendant friction force at the ball piston/cylinder wall contact would grow too large as speed increased. The design of the ball track impacted the magnitude of coriolis force somewhat, but it was not immediately apparent that track design could completely eliminate the net tangential force.

The mechanical dynamics of the design are conceptually simple, based on the 2-D equations of motion of an individual ball piston. Using Figure 4, assuming constant rotor rotation rate and simple Coulomb friction at the ball piston/cylinder wall contact, the three equations of motion are

$$\sum F_r = ma_r = F_P - \mu F_\theta - F_\rho \cos\psi + T\sin\psi$$
$$\sum F_t = ma_t = -F_\theta + F_\rho \sin\psi + T\cos\psi \qquad (1)$$
$$\sum M_G = I_G\alpha = -\mu F_\theta \rho - Tr$$

where the ball accelerations are

$$a_t = 2\dot{R}\omega + R\dot{\omega}$$
$$a_r = \ddot{R} - R\omega^2 \qquad (2)$$

and F_P is pressure force, F_θ is tangential contact force, $F_R = \mu F_\theta$ is friction force, $\omega = d\theta/dt$, $\alpha = d\Omega/dt$ (Ω is ball spin rate), R is ball position radius, r is rolling radius, μ is friction coefficient, m is ball mass, I_G is ball moment of inertia, ρ is ball radius, and ψ is track slope relative to tangential. All kinematic quantities, including ψ, are known if rolling is assumed, so the three problem unknowns are F_θ, F_ρ, and T.

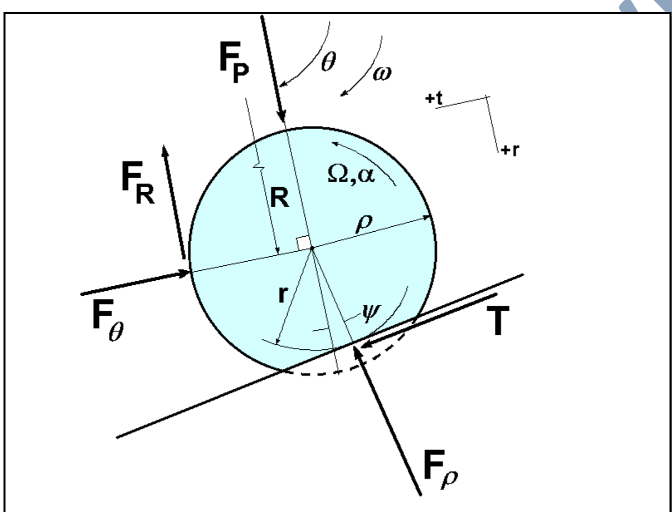

Figure 4. Ball piston free body diagram for power and intake strokes (ball position radius R increasing, and θ taken as zero at TDC before intake)

One must be careful to keep sign conventions and direction of non-conservative friction forces correct while considering all phases of the engine cycle, and one reaches the important result of tangential force on the ball and imparted to the rotor in the clockwise sense,

$$F_\theta = \frac{F_P \sin\psi - ma_r \sin\psi - ma_t \cos\psi - I_G \dfrac{\alpha}{r}}{\left\{\cos\psi + \mu \sin\psi + \dfrac{k\mu\rho}{r}\right\}} \qquad (3)$$

where $k = +1$ if $F_\theta > 0$ and
$k = -1$ if $F_\theta < 0$.

For reasonable values of μ, the denominator of equation (3) is always positive, so the sign of F_θ can be determined from the numerator alone.

Earliest designs not based on engineering analysis used a dual contact track with maximum rolling radius (equal to ball radius) at TDC, changing in approximately sinusoidal manner to a small rolling radius at BDC. This design allowed for maximizing stroke and maximum compactness. In that case, coriolis forces and attendant frictional losses would negate the useful power from combustion/ expansion at undesirably low rotor rpm.

Then sensitivity analysis of ball track design was studied using simple basic track geometry, i.e. sinusoidal variation of ball radius with rotation angle. It was thought that substantial reductions of inertial contributions to F_θ were achievable by “reversing” the track design so that full rolling radius was at BDC and a smaller rolling radius was reached at TDC, using a dual contact track. This approach was based on maintaining constant ball spin rate, which was thought to minimize inertial loads, and it was recognized that there would be some loss of stroke due to the track at TDC. It was found, however, that results were not much better, because of large coriolis forces that still existed. Figure 5 shows the individual contributors to rotor tangential force for an example of the constant ball spin rate track design. It is seen that the power producing force from combustion is dwarfed by the inertial loads, particularly the coriolis contribution.

Then sensitivity to rolling radius magnitude change was investigated by trial and error, and it was found that large improvements could be made by imposing a certain amount of ball angular acceleration in the proper direction to cancel coriolis forces. Figure 6 shows a comparison of net tangential forces for the simple constant ball spin rate track and optimized sinusoidal track. Inertial forces were decreased by almost an order of magnitude by this approach. The remaining force has about double the frequency, due to nonlinear ball track slope details that were not correctable by a simple sinusoidal track design.

Looking in more detail at equation (3), it is seen that along with the power producing contribution of F_P, there are also tangential acceleration forces from both translation and rotation of the ball. We can take these contributions together and minimize them by using track rolling radius to impose ball angular acceleration α. We can define the inertial load we wish to eliminate by

$$F_l = ma_r \sin\psi + ma_t \cos\psi + I_G \frac{\alpha}{r}$$

$$F_l = m(\ddot{R} - R\omega^2)\sin\psi + m\left(2\dot{R}\omega + R\dot{\omega}\right)\cos\psi + I_G \frac{\alpha}{r}$$

(4)

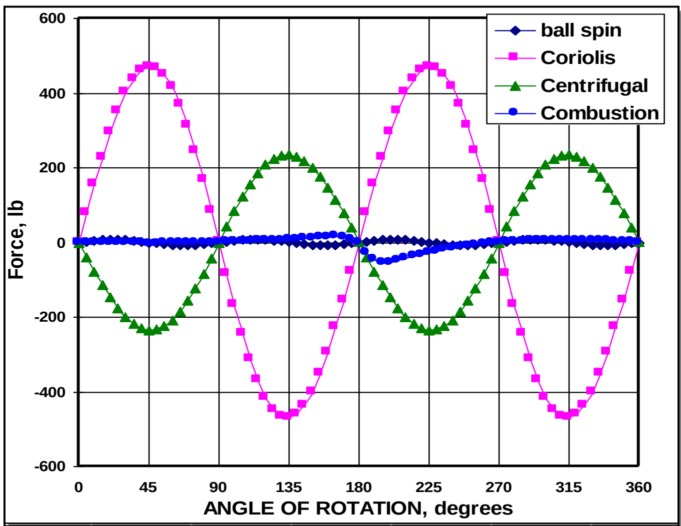

Figure 5. Individual contributions to ball tangential force for constant ball spin rate track (2 inch diameter silicon nitride ball, mean ball position radius=10.00 inch, 0.1 coefficient of friction, 5000 rpm)

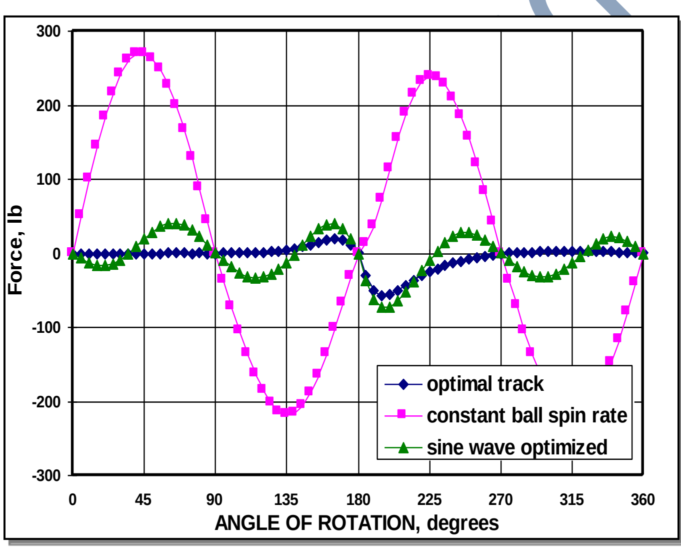

Figure 6. Net ball tangential force comparison for track designs (2 inch diameter silicon nitride ball, mean ball position radius=10.00 inch, 0.1 coefficient of friction, 5000 rpm)

Now, $\dot{\omega}$ is zero for constant speed operation, $R, r, and\, \psi$ are dependent only on θ, and $\dot{R}, \ddot{R}, and\, \alpha$ are dependent only on θ and spin rate ω due to the constraint of rolling. For example, the ball spin rate is

$$\Omega = \left\{ \frac{R(\theta)}{r(\theta)\cos\psi(\theta)} \right\} \omega \qquad (5)$$

Then differentiating with respect to time, the angular acceleration can be shown to be a separable function of θ and ω,

$$\alpha = \alpha(\theta)\omega^2 \qquad (6)$$

Similarly, all other time derivatives can be separated, and using primes to denote derivatives with respect to θ, one obtains

$$F_I = \left\{ m(R''(\theta) - R(\theta))\sin\psi(\theta) + 2mR'(\theta)\cos\psi(\theta) + I_G \frac{\alpha(\theta)}{r(\theta)} \right\} \omega^2 \qquad (7)$$

Thus, it is seen that for any rpm (ω), the geometry of the ball track (ball position radius *R* and rolling radius *r* as a function of rotation angle of the rotor) can be tailored to give exactly zero net force, by playing the ball angular acceleration against the ball translational acceleration. Given $R(\theta)$, $r(\theta)$, and ω, $\psi(\theta)$ and $\alpha(\theta)$ can be fully computed. Using a dual contact track, allowing the ball rolling radius to change adds the degree of freedom necessary to achieve this balance. Figure 6 shows, for the "optimal" case, how inertial tangential forces are completely eliminated, leaving only the combustion force to provide usable power.

It is important to point out that the resulting design is not a perpetual motion machine. The translational and rotational kinetic energy is simply exchanged in a prescribed manner to achieve the desired effect. In total absence of friction and other losses, the ball would roll around the track in perfect synchronization with the constant speed rotor without tangential interaction forces.

It is difficult to solve for the optimal geometry of the track explicitly, due to the trigonometric complexity of the governing equation (7). Iterative numerical methods, such as Newton Raphson, can be implemented to solve for the ball rolling radius, given a functional form for ball position radius. A logical assumption for $R(\theta)$ is sinusoidal, but a different form useful for engine cycle optimization is just as easily used in the computation of $r(\theta)$. The track slope $\psi(\theta)$ depends completely on $R(\theta)$ by the equation

$$\psi = \tan^{-1}\left\{-\frac{1}{R(\theta)}\frac{dR(\theta)}{d\theta}\right\} \qquad (8)$$

so maintaining zero net force in equation (7) consists of solving a nonlinear transcendental equation for $r(\theta)$ at discrete values of θ. Figure 7 shows an example of the optimal ball rolling radius variation with rotation angle for a 2.0 inch diameter ball with a mean ball position radius of 10.00 inches, and sinusoidal $R(\theta)$. Using the pure sine wave comparison in Figure 7, the form of $r(\theta)$ is seen to be nearly sinusoidal, but there are small nonlinearities introduced by track slope effects. Nevertheless, the track is readily producable using computer controlled machine tools.

Note that the minimum rolling radius for this case is 0.81ρ at TDC, so a portion of the stroke available, 0.19ρ, is lost. One must iteratively choose a stroke, implicit in the definition of $R(\theta)$, and then check whether it is geometrically feasible for rolling radius at the end of the computation. Figure 8 shows the lost stroke as a function of ball size and ball position radius. Larger balls and ball track radii are better for minimizing stroke loss. Figure 9 shows minimum rotor radius as a function of ball size, based on a reasonable stroke loss of 25%. Less stroke loss can be achieved by using larger rotors, but there will be a practical design trade-off against centrifugal loads and engine size.

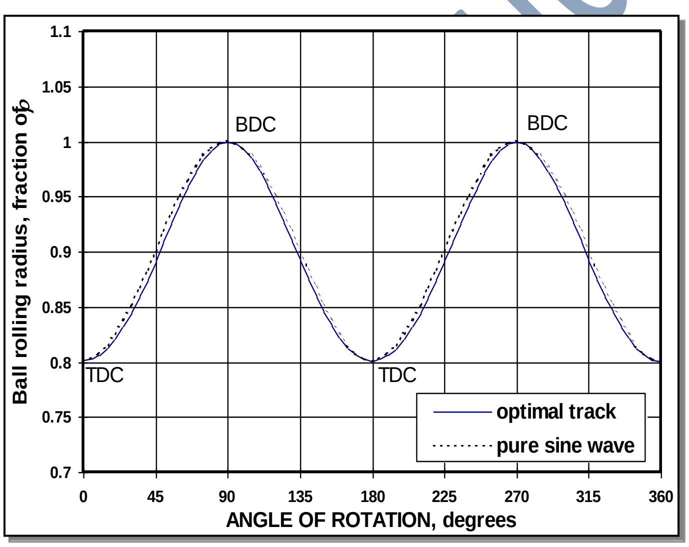

Figure 7. Optimal track rolling radius compared to pure sine wave (2 inch diameter ball, mean ball position radius=10.00 inch)

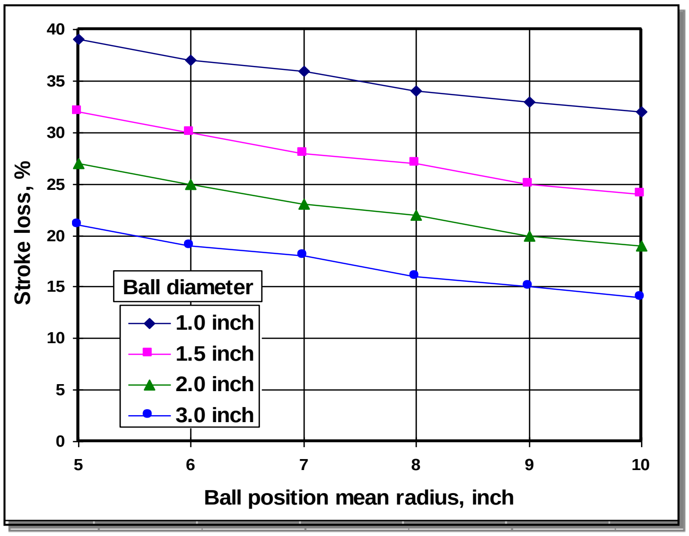

FIGURE 8. STROKE LOSS AS A FUNCTION OF ENGINE DESIGN

ENGINE PERFORMANCE PREDICTIONS

Simulation Model - A multi-energy domain engine simulation model was developed for efficiency studies. The model was based on the equations of motion (1). Approximate models for combustion kinetics, steady state heat transfer, working gas thermodynamics, Coulomb friction, and ball piston leakage were included.

Leakage modeling was based on simple orifice flow neglecting ball spin, with choked flow occurring at sufficiently high pressure ratios. An orifice coefficient C_d of 1.0 was used for conservatism, and for lack of available data. Leakage at the rotor/stator bearing was assumed zero, because bearing calculations indicated leakage could be controlled very well by altering rotor width (and thus bearing land width).

Combustion kinetics was simulated by a simple time lag for linear pressure rise to a level based on constant volume stoichiometric steady state combustion of gasoline (octane and air). Working gas thermodynamics was based on ideal gas laws with heat transfer. Steady state heat transfer was based on approximations of conduction and convection between working gas, ball piston, and cylinder/rotor, with cool intake air flow over the rotor exterior and the ball exposed outer hemispherical surface.

The model was simulated at constant rotation rate, simulating an engine load with substantial inertia. Output shaft torque per ball piston was the main output quantity, and also internal forces, pressures, and temperatures were output for review. The model was executed in a matrix mathematics program called Gauss [6].

Simulation Results **-** The four stroke rotor design was the main configuration of interest. The simulation model was exercised for a wide variety of cases considering different ball size, rotor size, leakage and heat transfer assumptions, and rpm. The optimized track design already discussed tended to narrow interest to larger balls, however, and that is the data to be presented.

Figure 10 shows the specific power curves for the constant ball spin rate and optimal track cases (2 inch ball diameter, mean ball position radius=10.00 inch, 0.10 coefficient of friction). They are compared with a case of no friction, leakage, or thermal losses (but adiabatic pumping and estimated combustion loss is included). It can be seen how important the "inertial cancellation" of optimal track design really is. The constant ball spin rate power curve drops quickly as rpm reaches usable range due to inertial force growth. With the optimal track, the power curve is essentially linear (other factors may reduce power at high rpm, such as engine flow limitations).

Figure 11 shows engine torque for the same cases, and the influence of leakage can be more readily seen at low rpm, where torque drops substantially below 1000 rpm. Above 1000 rpm, efficiency of about 60% is controlled by friction and thermal loss. Of the 40% loss, 20% is friction loss, 18% is thermal loss, and 2% is leakage. Leakage decreases with increasing speed, so efficiency increases slightly with speed.

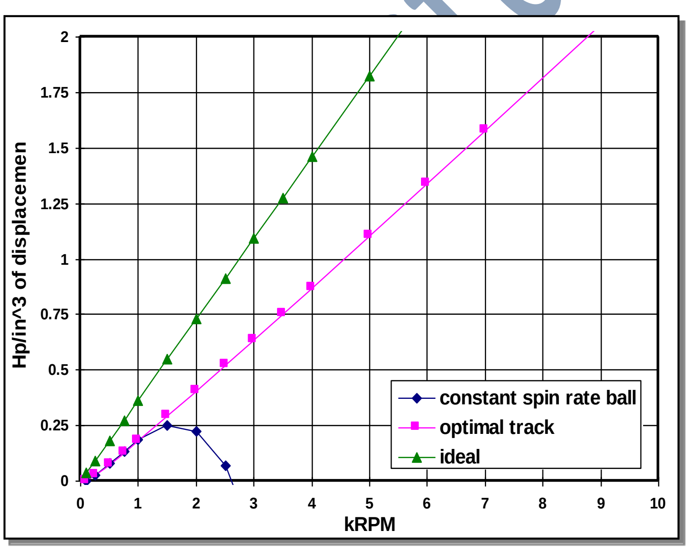

Figure 10. Specific power comparison for track designs (2 inch diameter silicon nitride ball, mean ball position radius=10.00 inch, 0.1 coefficient of friction, ball diametral clearance of 0.001 inch)

In comparison, typical losses for water cooled spark ignition engines [7] are 50-55%, of which about half is friction and half is thermal, with negligible leakage. The thermal losses have been greatly decreased in the new design by elimination of heat transfer to a water cooling system.

Design Choices - For the example engine, steady state temperatures were estimated as 700°F for the cylinder/rotor and 2400°F for the ball piston. To sustain that temperature, silicon nitride is chosen for the ball piston. Silicon nitride is also a good choice for light weight (lower centrifugal forces) and low friction, as well as low coefficient of thermal expansion.

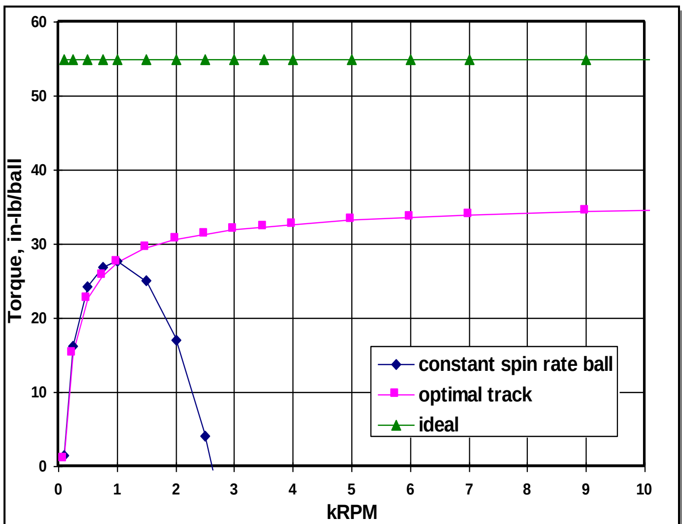

Figure 11. Torque comparison for track designs (2 inch diameter silicon nitride ball, mean ball position radius=10.00 inch, 0.1 coefficient of friction, ball diametral clearance of 0.001 inch)

With a silicon nitride ball piston and steel cylinder/rotor, which have widely different coefficients of thermal expansion, but also widely different steady state temperatures, the thermal expansion is almost perfectly matched. Thus, the material selection has a secondary benefit of maintaining operating clearance within 10-20% of nominal over a wide range of engine operating temperatures. In an actual engine development the thermal expansion can be tuned by rotor external design for cooling (cooling fins or outer rotor width, for example).

The use of a silicon nitride cylinder wall was considered, but friction of like ceramic materials is generally high. Research results concerning special silicon nitride compounds may be useful in production, however [8].

Note that it may be beneficial to introduce active lubrication into the engine. If friction can be reduced from 0.10 to 0.05, engine efficiency can be increased from 60% to 70%. There are trade-offs to be considered with active lubrication, including residue accumulation, emissions, and maintenance. One reasonable approach would be oil jet spray into the local cylinder wall contact area from the outside of the rotor, or oil pickup by the ball itself from the track area just before the power stroke.

ADVANTAGES

- The stroke magnitude and rate can be different for different stroke in cycle so that it provides the possibility of converting more energy to the shaft power by greater expansion during the power stroke.
- It has ability (i.e. in multi energy domain engine) to complete any even numbers of strokes per revolution in single rotation of rotor
- There is no much moving parts in the ball piston engine thus the power out put at the shaft is high.
- In this engine the frictional losses are low and independent of operating speed in compare to conventional piston engine.

DISADVANTAGES

- Flow is choked during combustion due to high pressure differential and small clearance area
- The friction and wear at the ball piston/cylinder wall sliding interface
- Leakage through the ball piston/cylinder gap is a significant factor for engine efficiency at low speed

APLICATIONS

- It can be applied to compressor.
- The multi cylinder ball piston engine can be applied to pump ,motor.
- It can be applied to engine.
- The wankle advanced two stroke ball piston engine can be applied to land mover, standard motor cycle and car and also for racing cars

CONCLUSIONS

Analyses based on the design assumptions showed that the ball piston engine has potential for achieving higher efficiency than piston internal combustion engines. In addition, subscale tests have shown that critical leakage and friction characteristics are consistent with design assumptions. Thus, the feasibility of this new engine concept based on ball pistons has been proven.

A new approach to kinematic design has been devised to eliminate friction contributions from inertial forces in the engine. On the other hand, conventional carburetion/induction and exhaust systems are applicable to the new engine.

Some material problems were encountered in subscale testing, indicating that more detailed material selection was warranted. The material selection has been done in anticipation of additional subscale tests to extend the range of speed and duration of simulated operation. Baseline material for testing is M2 tool steel.

Shortly after cylinder material selection is verified in subscale tests, fabrication and testing of a prototype engine will be undertaken. The prototype will be used to finalize design details such as thermal design, transient operation, starting, and cylinder wall treatments with actual combustion environment.

The new design concept can be immediately applied to compressor and pump applications in parallel with further engine development. The concept holds immediate promise for high efficiency and low cost in these applications, where temperatures and loads are more benign and lower cost materials can be used.

REFERENCES

- www.google.com
- www.wikipedia.com
- www.studymafia.org
- www.projectsreports.org