The FMV story

Below is an article written by Rob Metkemeyer and Enrico Flores in the winter of '78-79.
It's about their FMV team race engine.

F stands for Enrico Flores
M stands for Rob Metkemeijer
V stands for Hans Visser

Picture made in Pienne 2001

This is the original version with hand drawn pictures

The F.M.V.

or a few ideas on model engine construction.

Rob Metkemeijer and Enrico Flores.

1. Introduction.

The design and construction of the FMV took place from November 1976 until right before the 1978 World Championships and must be considered as a synthesis of ideas we took from quite a few years of teamrace-experience and espionage.
Flying experience taught us what we needed: A fairly powerful and economical engine and, far more important, one that would GO under all circumstances.

A teamrace engine is only worth for the number of laps it can run at top speed, and its capability to withstand wrong - especially lean - settings, and/or heavy races.

The once, and maybe still, superior Russian engines Onufrienko, Krasnorutsky, Maslov) were never more powerful than good Bugls or Rossi's. They were just built into excellent models and had an amazing capability to withstand any setting in any type of race.

Some Bugls MK 1 and MK 11 and many Rossi's were probably more powerful, yet generally incapable of completing a race if set for maximum speed and economy. because very prone to sudden seizures.
So a "good" setting on these engines was a safe one, slightly rich and undercompressed, but slower with possible problems of range. With the "discovery" of the Nelson l5D motor in the second half of 1977 it was clearly shown that contests could be won with an engine not any faster than the opposition but far more consistent. Its consistency in a race, rather than its power, impressed us deeply and we realized that Henry Nelson was on the right path. We must admit that it wasn't very easy to find out the reason of its superiority. Moreover we had already lost two years, because in 1975 we had been so stupid as to judge this engine only by its appearance.

2. Distinctive features of a teamrace engine

A teamrace engine, is the only type of competition model-engine where power as well as fuel consumption matter. As everybody knows, diesel engines have a good reputation for economy, and that is why they are used in this racing class. Engines with with a low fuel consumption get very little internal cooling from the fuel, so they are at the mercy of cooling from outside, air-cooling because of weight.
In all other types of competition engines small mechanical imperfection's are reckoned with by simply opening up the needle, and therefore never recognized. In TR-engines this solution is not acceptable. However, even the best TR-engines are unfavorably with the big diesels with respect to specific fuel consumption.
As every TR-mechanic knows there are settings with more compression and less fuel which make the engine running faster for a short while ...... and then a sudden stop follows caused by rise of internal temperature and/or friction and consequent pre-ignition. Because in competition flying the running conditions of TR-engines are very close to this limit we are convinced that TR-engines can teach us a great deal more than any other type of engine about the existing mechanical and thermal problems.
This, and the following articles, will describe general mechanical principles and thermal considerations on model engine construction and the way they were applied to the FMV. Hereby we trust that persons concerned with model engines achieve a deeper understanding of them and will hopefu11y become able to recognize and so1ve the problems they encounter.
Our approach can be summarized as follows:

- Prevent local sources of friction, and thus heat. (Excessive friction and heat are generally generated by vibration, wrong tolerances or deformation, wrong material combinations and lack of lubrication).
- Give cooling and lubrication to those places where necessarily heat is generated: combustion chamber, small end, ball races, piston and cylinder.
- Try to stabilize the system in such a way that, at rising temperature, friction in all moving parts does not increase (right material choice with regard to thermal expansion coefficient is extremely important. If by thermal deformation friction is increased, temperature will locally rise more, giving more thermal deformation etc. and the engines seizes).

With the limitations in all existing TR-diesels on the mechanical side, we think it is absolutely useless to do a lot of research by means of improved gasflow, as long as too much fresh gas is not wasted during the filling of the cylinder. Quite a few experiments on cylinder timings and the shaping of ports taught us that you'll find all kinds of differences, but never a significant change in <u>efficiency</u> of the motor. The reason being that timing is a factor of power more than a factor of efficiency. The only wrong things was decreasing the differences between transfer and exhaust timing in the cylinder to less than 18. For a Schnuerle scavenging layout all exhaust timings between 125 and 150 and many different portshapes seem to work. Rotary vales opening at 45 15 A.B.D.C. and closing 50 15 A.T.D.C., worked as well Experiments showed, that it is always possible to compensate for the above mentioned variations by changing propeller and/or venturi.

3. The bottom end.

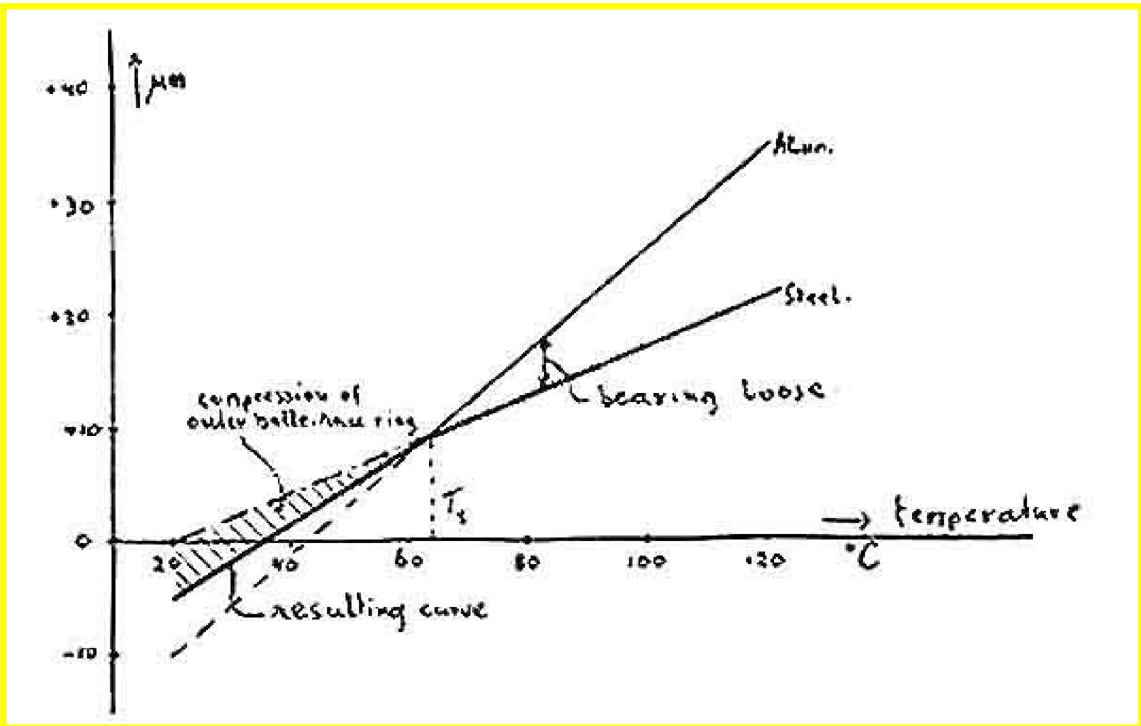
The bottom end of the motor (crankcase, crankshaft & bearings and induction system) forms the basis of the engine, and a look at most commercial engines shows that this part is largely underestimated. In team race however the necessity of improved constructions is clear, because motor are simply falling apart there.

3.1. Crankcase and bearings.

One of the most serious problems in conventional motors with ball bearings in an aluminium case, is that because of the different expansion coefficients of thermal expansion between aluminium and steel, the fit of the outer bearing ring in the case gets looser at rising temperature. The order of magnitude of this effect is 0.015 mm every 75 °C of temperature rise. (see fig. 1a and 1b)

Fig. 1 a.

Thermal expansion aluminium housing vs. steel outer bearing ring.
Aluminium housing made 0.01 mm smaller than bearing).D. at 20 °C.
(Wall thickness steel ring = 1.3 mm, wall thickness aluminium housing = 4 mm)
At 20 °C, the ballrace outer ring is compressed 0.005 mm.

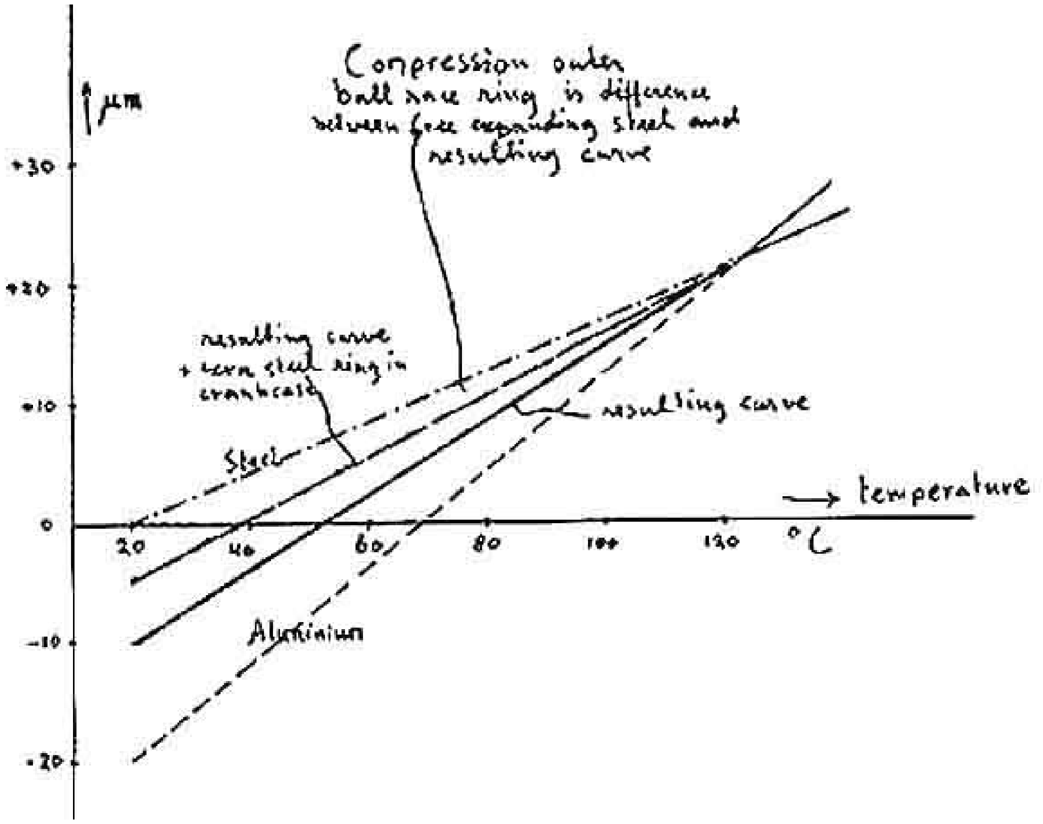


--- free expansion aluminium housing
-.- free expansion steel bearing ring
___ resulting curve bearing outside diameter and diameter aluminium housing (bearing in housing)

Temperature below 65 °C, then bearing has an interference fit. <u>Temperature over 65 °C, bearing comes loose.</u>

Fig. 1b

As fig. 1a, but aluminium house made 0.02 mm smaller than bearing O.D. at 20 °C.
At 20 °C, the ballrace outer ring is applied, the relative stiffness of the outer ball race ring is increased and the stiffness of the aluminium housing is decreased.
At 20 °C, the ball race outer ring is compressed 0,005 mm, like in fig. 1a, but temperature range is extended up to 120 °C



--- free expansion aluminium housing
-.- free expansion steel bearing ring
___ resulting curve bearing outside diameter and diameter aluminium housing (bearing in housing)
~~~resulting curve if outer bearing ring is stiffened with an extra ring, 1.5 mm wall thickness. (curves applies to O.D. extra steel ring)

The constant "hammering" and a turning moment will thus make the bearing outer ring creep around, ruining the housing. Before this happens, the strong influence of mechanical load on engine temperature will produce the very well known popping run followed by a sudden seizure.
Many TR-flyers recognized this problem and quite a few solutions were seen.
Jim Plaunt, for instance, used three set screws to prevent the rear race from moving in his 1976 Rossi RV.
A same type of solution was seen in one of Maslov's engines. He used a 0,6 mm diam. key between bearing and house. Both solutions affect the roundness and/or stiffness of the outer ring and must be considered as not ideal.(See fig. 2a and 2b). A far better solution is to Loctite the bearing at 100°C into the crankcase (use Loctite bearing fit with activator and do it quickly!). This method was quite widely used by us in MK I and MK II Bugls, and usually lengthened the life of an engine for a while. The only problem with this method comes if you must change bearings. Some Russian engines showed steel rings shrunk and maybe also glued into the crankcase. This system at least solves the above mentioned problem of replacement of bearings, but the combination of aluminium case and a shrunk-in steel ring has a resulting thermal expansion coefficient somewhere in between aluminium and steel, so the problem is only half solved (see fig 2c and 1b).
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FIG.2 SYSTEMS FOR PREVENTING BEARING ROTATION

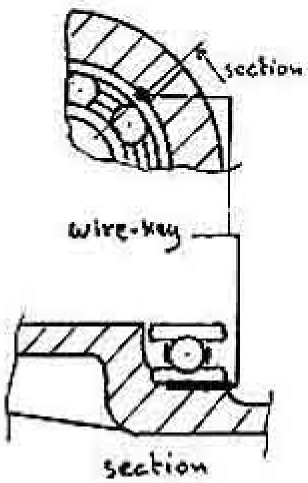


fig 2a
Maslov's solution

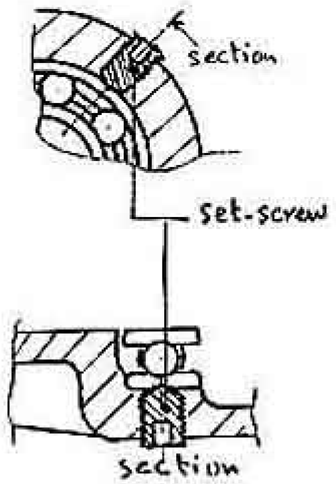


fig 2b
Plaunt's solution

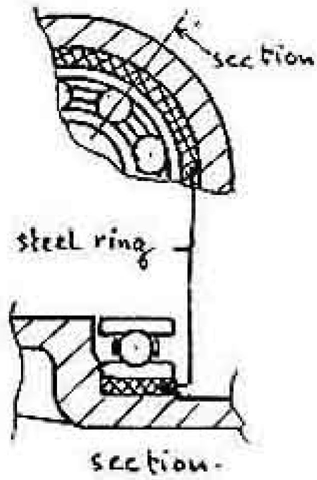
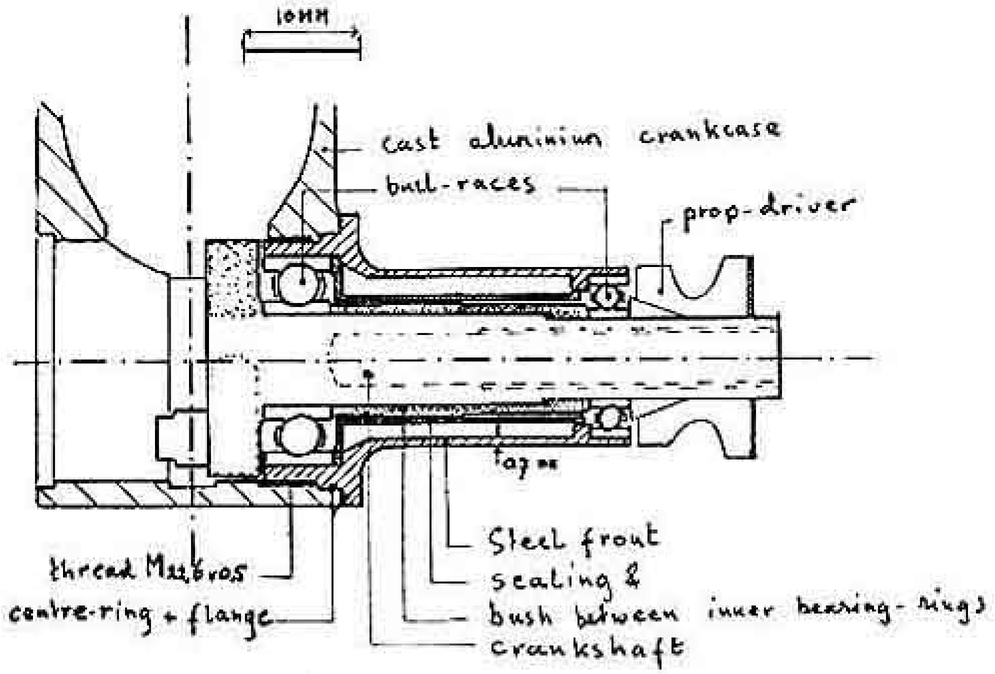


fig 2c
steel in crankcase

One of the best solutions to the problem is from Henry Nelson. He uses simply a very strong interference fit (~0.025 mm), which will stand temperatures up to ~150 °C before the bearing comes loose. Normally crankcase temperatures will be under 100°C. A few conditions have to be fulfilled.
Firstly the material of the crankcase has to be able to stand a relative high stress at room temperature.
Secondly, bearings of a special high clearance type are needed to compensate for the compression of the outer bearing ring at lower temperatures. At higher temperatures, the bearings will run with quite some play, both radial an axial, which won't necessarily be a big disadvantage, up to a point.
However, replacing bearings at room temperature will give slightly decreasing interference fits, and asks for special tools. Replacing them at temperatures above 110°C - 150°C, which is necessary to do it without applying force, will ruin bearings. (advice: let Henry do it: he knows how it's got to be done).
The solution that we think is best was firstly shown by Krasnorutsky: the all steel front housing.
We tried them first in MK I Bugls and the effects were quite amazing compared with the (worn out) original aluminium front housing: no sudden and unexpected seizures at healthy sounding settings anymore! Fig. 3 shows the principles of the FMV front housing, a screwed-in type, developed after a good look at Krasnorutsky's 1977 engine.

The front housing is Loctited (Loc-tite hot retaining compound + activator) at 50°C into the crankcase, which means that above 50°C no change in interference fit of the main ball-race exists.
The interference fit of the front race is not affected by rising temperature of the engine. The fit we use is about 0.001 - 0.003 mm of interference for the rear race and 0.000 - 0.001 mm for the front race. More interference means loss of play in the bearing. This means they just can be put in with a (strong) human thumb or soft hammering with a piece of wood.

fig 3



The FMV steel front housing

Getting the ballrace housings to size is an extremely accurate job, not helped by our deformable, normal clearance, bearings. With high clearance bearings more interference could be used and the relative accuracy of the house becomes less critical, but we still can't get these bearings in our sizes (8x19x6 and 7x14x3,5) without ordering large numbers.

3.2 The bearings and the shaft.

Since both shaft and inner bearing ring are normally made out of steel, no thermal expansion problem occur. However it is of high importance that the inner ring is prevented from creeping, and wearing the shaft.
In all commercial engines the inner ballrace ring is retained on the shaft by a more or less heavy interference fit, or worse, by a sliding fit (Rossi, Super Tigre, K & B). A tight fit, like in the Nelson, makes putting the shaft in and out a specialists job (same advice as before!) and it also calls for high clearance bearings.
Bearings with normal clearance won't accept interference fits of more than 0.001 - 0.002 mm interference on the shaft without loosing their play. Sliding fits on highly stressed T.R. engines will cause all sorts of distortion and vibration problems, increasing friction and eventually ending up with a cold forged, oval shaft.

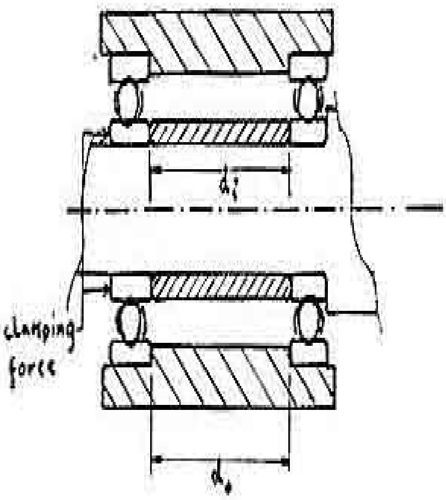
In quite a few cases we found remarkable, if temporary, improvement just Loctiting the shaft to the bearings!
In the FMV the construction shown in fig. 3 is used. A bush clamped between the inner rings of the bearings by tightening the prop nut will prevent both inner rings from turning. For this reason we can easily use a non-hardened shaft (material DIN 100Cr6, En 31 (G.B.), E 52100 (U.S.A.), the same as our front housing and bearings), with 0.001 - 0.002 mm interference fit on the bearings. This bush is quite common in "real" machinery and has been used for many years in Russian TR-motors. It can be adjusted to such a length that the bearings are set to give the desired axial play to the shaft.

3.3 Axial and radial play of the shaft.

Apart from turning the crankshaft can, and probably will, move (or better vibrate) in axial and radial direction. These vibrations are directly induced by the ignitions of the engine and are strongly influenced by the play of the ballraces. They will increase the friction, heat generation and wear of the bearings.

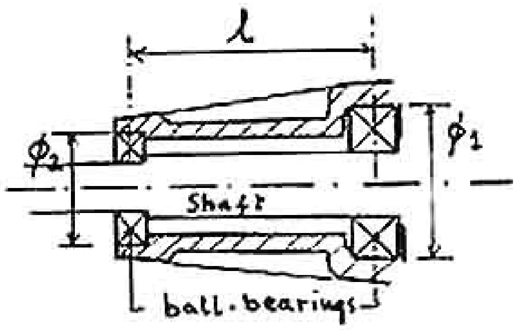
In ball bearing technique it is normal to control the play of bearings by "preloading" them. This means, that in the case of a shaft supported by two bearings, both axial and radial play are controlled by setting the inner and outer rings at slightly different distance (See fig. 4 and fig. 5).

fig. 4 Preloading ball bearing



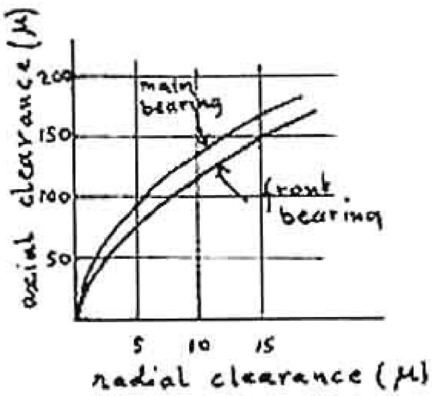
If di=do, axial and radial play is equal to that of one ball-bearing (~0,08 mm in our case)
If di is chosen smaller than do, axial and radial play decrease. If, in our case, di=do-0.08 no play is left

fig 5a



Shaft and bearings in an aluminium crankcase.

fig.5b



Curves of relationship between axial and radial clearance for Nelson bearings

fig.5 Axial and radial clearance of the bearing in an aluminium crankcase (example: Nelson)

Starting from zero clearance at T=20 Â°C we find from curves, similar to fig .1b and fig 5b, that at T=80 Â°C:
Ã†1 increases 4mm, so axial clearance will be 0.08 mm.
Ã†2 increases 3.5mm, so axial clearance will be 0.06 mm.
The different expansion of the crankcase (length l) and the shaft will preload the bearings for about 0.02 mm (for l=30 mm) over this temperature range.
An axial clearance of the shaft of 0.05 results.
Axial load in front direction will be taken by the main bearing, in rear direction by the front bearing.
A not "free" feeling shaft at 20 Â°C (zero clearance) is necessary in a Nelson to keep down clearances at running temperatures.

Our bearings when new, have an axial play of 0.06 - 0.08 mm.
We found this little too much, causing setting and overheating problems as well as fast wear of ball bearings. By shimming, we make the bush between the inner rings about 0.03 mm shorter than the distance between the outer rings, resulting in 0.03 - 0.05 mm axial play. This seems to be a good compromise and will stay constant over a large temperature range in an all steel set-up.

The reasons that no less play can be used safely, are the following:

- During running the inside of the motor (shaft) will always be warmer than the outside (front housing) simply because there's cold air outside and there's no other way for the generated heat to go. Therefore the inner ring is expanded and radial play (being around 0.003 mm in our case) will decrease by 0.001 mm with every 10Â° C rise of temperature difference.
  This effect, helped by a too well cooled front housing, committed quite some bearing ravages in our Rossi front-exhaust, the test-bed for many experiments with steel front housing. That's why in the FMV model direct cooling to the front housing is prevented by using a spinner (see fig. 16)
- Play is necessary to allow the shaft to bend during ignition. Calculations show that about 0.4Â° rotation at crank pin location occurs.

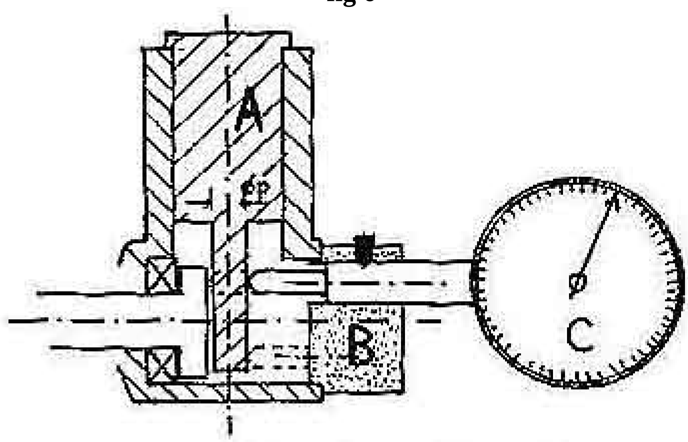
With an aluminium front housing controlling radial play by means of axial preloading is nearly impossible. This is because the bearing play induced by the radial expansion of the house will be much greater than the play reduction due to the lengthening of the house for a given temperature variation.

Setting a preload for running temperature will ruin the bearing when the engine is cold. In well fitted engines, like the Nelson, preloading is controlled by the interference fit of the bearings and the different expansion of the crankcase and the shaft, just giving the right radial and axial play at running temperature. (See fig.5).

3.4 Locating and sealing of the crankshaft, lubrication of the front bearing.

Because of the limited axial movement of both ends of the conrod it was essential that the middle of the crank pin was located exactly (within 0.01 mm) on the center line of the cylinder.
The measurements were carried out with the equipment drawn in fig. 6.
From fig. 6 it is clear that, by measuring the distance difference to the dummy con-rod with the dial gauge in top and bottom position, the perpendicularity between the shafts-axis and the cylinder axis can be checked with the required accuracy of about 0.05Â°
The axial play of the shaft can be checked with the dial gauge on the crankweb by pushing and pulling the shaft softly.

fig 6

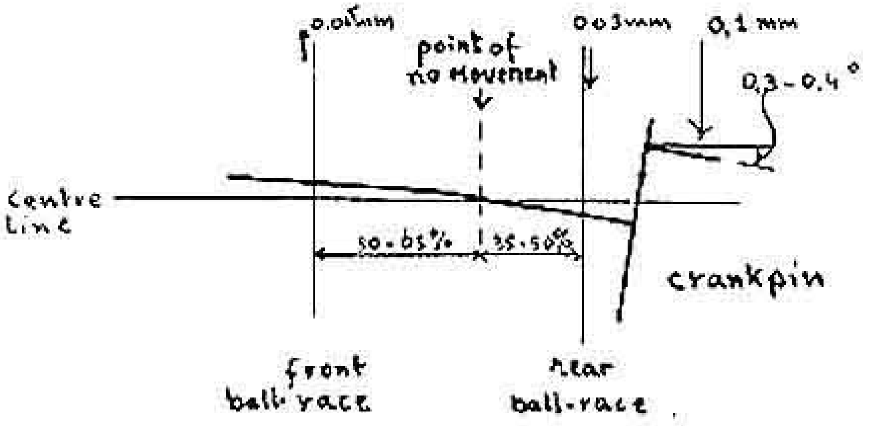


Measuring equipment to find the distance between the centre line of the cylinder and the rear plane of the crankweb. Part A fits closely in the cylinder hole and has a pin (Ã†p), being concentric with the cylinder. The pin is the "dummy conrod". The dial gauge , held in B, gives two readings, one for the crankweb and the second for the dummy conrod
The distance between the centre of the cylinder and the crankweb can be easily calculated, knowing Ã†p.

In the case of the FMV the position of the shaft is set by putting a separate part into the front housing (See fig. 3) The thickness of the rear ring determines the exact position of the shaft. This part also gives the sealing around the shaft.

A difficult point has always been the clearance necessary around the shaft to prevent touching. Since we want the clearance as small as possible to prevent excess fuel leaking through the front, calculations were made of the radial movement of the shaft at different places caused by the ignition of the motor.
Fig. 7 shows the approximate movements, for an 8 mm shaft and 8-19-6 mm and 7-14-3,5 mm bearings.
With a stronger shaft and bearings like the Nelson and Bugl have, these movements will be slightly less, but still of the same order of magnitude.

fig. 7



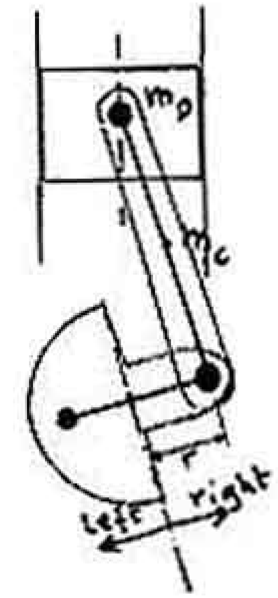
Radial movements of the shaft during ignition. Figures are not exact because peak pressures in combustion chamber are not known exactly. Point of no movement is independent of actual forces.
(Note that a bell valve fixed to the crank pin will move approx. 0.15 mm)

In any case it will be clear, that the best point of sealing the shaft is between the rear 1/3 and half the ball bearing center distance, where the movements are smallest. This is unlike most commercial RV-engines having the sealing right in front of the main bearing.
A clearance of 0.06 - 0.08 mm in diameter between the sealing ring and the bush around the shaft over a length of about 10 mm has proved to prevent any touching and gives barely enough <u>leakage</u> to keep the front bearing wet (1 or 2 drops per tank).
This lubrication is forced by average crankcase pressure through the small gap around the shaft.
In most other engines this guidance of lubrication doesn't exist, because after passing the sealing there's no controlled way for the fuel to reach the front bearing.

3.5 Balancing.

Since we always found that vibrations (think of a not fully tightened pan), cause losses in performance, economy and consistency, we are convinced that keeping the dynamic forces, (the cause of vibrations) down to minimum is one of the ways to make a better engine.
A great deal is known about balancing, one of the things being that it is impossible to balance a single cylinder engine completely, without bringing the reciprocating masses down to zero (except by using auxiliary rotating counterweight). One logical way of improvement is to lighten piston and con-rod as much as possible.
Is there any other reason, that aluminium pistons work so well??
The normal way to "balance" single cylinder two-stroke racing engines is to counterbalance the shaft in such a way that first order horizontal and vertical reciprocating forces are equally strong, which means that the dynamical first order forces are minimized. (See fig 8 for explanation.)

fig. 8



Unbalance in a 1 cylinder engine

Minimizing total unbalance forces of first order is done by making horizontal and vertical components of unbalance equal, in formula.

Mass moment left crankweb - mass moment right side crankweb (incl. crank pin)
$-r*0.5\ m_c= 0.5\ *r*(m_p+0.5m_c)$

m_p = mass piston
m_c = mass conrod
r = 0.5 stroke

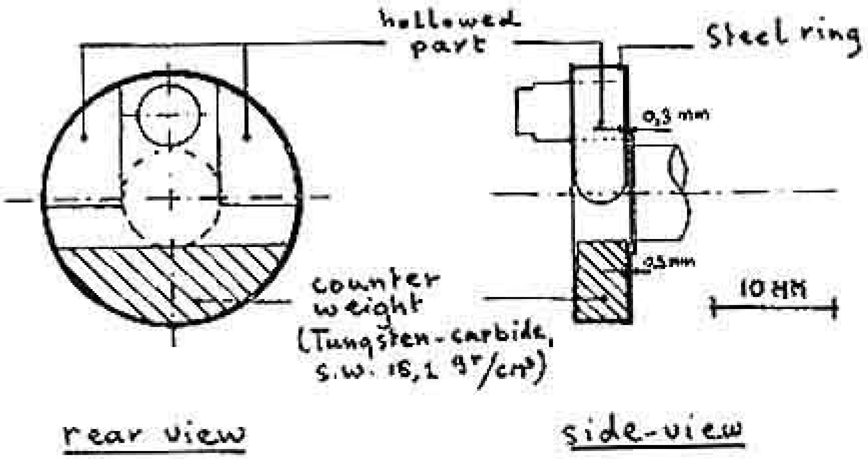
In the presently marketed 2,5 cc engines with iron pistons only 15 - 20% of the piston's weight is balanced.
In ABC or AAC engines this is up to 30 - 35%.

To get the FMV up to 50% balance with a Meehanite piston two things were necessary:

firstly the piston weight should go down as much as possible, secondly a counterweight was put into the crankshaft. The piston construction is to be described later, the hollowing and filling of the crankweb is drawn in fig. 9. The counterweight is made out of tungsten-carbide with a density of 15.1 gram/cm^3. (about 1.5 times that of lead).

A standard type cast iron piston, including pin, about 2 grams heavier than ours, would have made a counterweight out of gold or platinum necessary. Financial problems then forced us to design a super light piston. With AAC piston/liners expected to come this year, the problem of balancing will be solved more easily and with better result, with only a small counter weight in the shaft. (By the way, the FMV shaft with counterweight and bushing is as heavy as a standard Rossi RV shaft).

fig 9



Crankweb of the F.M.V.

3.6 The big end. The heel of Achilles of the motor.

Induction system

The only real mistake in the design of the FMV was (and still is) the size of the crankpin: 5 mm dia times 3 mm length. The reason to choose a short pin was to minimize the effect of non perpendicularity caused by crankshaft bending (during ignition up to ~0,4° rotation) and crankcase-bending due to temperature difference between front and rear of the cylinder (approximately 20° C, giving about 0.03° forward bending). With a short pin a small crankcase volume is obtained which is generally considered to be an advantage in a two stroke engine.

Unfortunately because of the relatively small projected area (15 mm2) compared with other motors (Rossi 4,5 x 4,5; Nelson 4 x 4,7; Super Tigre 5 x 4,5; Bugl 6 x 4,5) the generated heat per mm^2 is relatively high. (The specific generated heat is proportional to r.p.m., pin diameter, pressure, and coefficient of friction).

However, considering the small difference in area, it was difficult to justify why the FMV turned crankpins blue and ruined big-end bushes in the con-rod, while the Nelson kept their crankpins and con-rods looking brand new even after hours of running. It must be said that the first ruined con-rod we saw in a Nelson (King-Rudd's) brought us the answer. King-Rudd had changed the induction system to a Natalenko type, having the same disadvantage as the Russian bell-valve type we used at the time, to direct no fresh gas flow to the big-end. An easy solution seemed to make a disc-valve on ballraces, admitting the fresh gases right in the bottom of the engine. But surprise-surprise, no improvement was found.

So at last we tried the Nelson (K&B) rotor, and it solved all our big-end problems. The explanation is quite simple after all. In a disc-valve all the heavy parts (droplets of fuel) in the fuel-air mixture were centrifuged to the outside without hitting the big-end. In the case of the K&B Nelson-rotor the particles will hit the crankpin on their way, also guided by the groove in the rotor front, cooling and lubricating the pin.
The big-end of a motor really being a "hot spot", could be used to help vaporizing the fuel that arrives there, thus giving more efficient combustion (hope, hope!!).

At Woodvale, we had the opportunity to talk to Don Jehlik (World Champ. T.R. '66 and '68), more or less the supposed "inventor" of this induction system used in his ETA and HP and it turned out that he had always known that this really helped to make those motors superior to everything flying around at the time.

A few things became clear now:

- Bugl had to go to a 6 x 4,5 mm crankpin to solve his problem;
- Rossi RV's give more problem than Rossi FI's, especially at the big-end side;
- Our Rossi front-exhaust stopped to be a 50-lapper after changing to a Natalenko drum.

The gas-flow of the Nelson-K & B rotor is identical to that of a front induction system, so why not use an FI-engine in TR? We think there are four main reasons not to do so:

- The crankshaft on a FI-engine is a small monster, stiffness and strengthwise.
- It makes the use of a bush between the inner rings of front and rear bearing nearly impossible.;
- The carburettor will probably be exposed to turbulent air all the time and also to dust while on the ground;
- A too long distance between tank and spray-bar will cause acceleration troubles and fuel pressure differences between flying and ground running, depending on the position of the fuselage.

Getting back to the size of the big-end, there are indications that a length-diameter ratio around 1, like Rossi and Super Tigre have, is a good compromise between sensitivity for the shaft's oscillation and that for end effects of an excessively short pin (pressing oil to the sides and shortening the length of the lubricated part considerably). Maybe this effect made the FMV big-end extra critical for the type of induction systems.

A few words about the material of the pin. For still difficult to explain reasons the <u>hardness</u> of the pin has turned out to be of extreme importance. The needle bearing rollers used in Nelsons, the FMV's and in some other motors are very hard (over 1000 Vickers), which is more than any other crankpin in one-piece crankshafts.
After the Worldchamps and a very interesting talk to Don Jehlik we plated with hardchromium the pin of one FMV (layer thickness 0.001 mm) and found about 0.5 secs/10 laps improvement.
A hard chromium plated surface, being extremely hard and also porous seems to have significant advantage.
Be sure that the hardness of the chromium plating is at least 1000 Vickers. For example, the chromium plating on S.T. G15 and X15 big-end pins is far too soft to be used with a hard type bronze or cast iron bush in the con-rod, and fell off during a few experiments we did.
Since we really believe that a lot of power of our motors is spoiled in the big-end, our future investigations will be focused on materials selection and design of this part, including improved cooling and lubrication.

4. The top end of the motor.

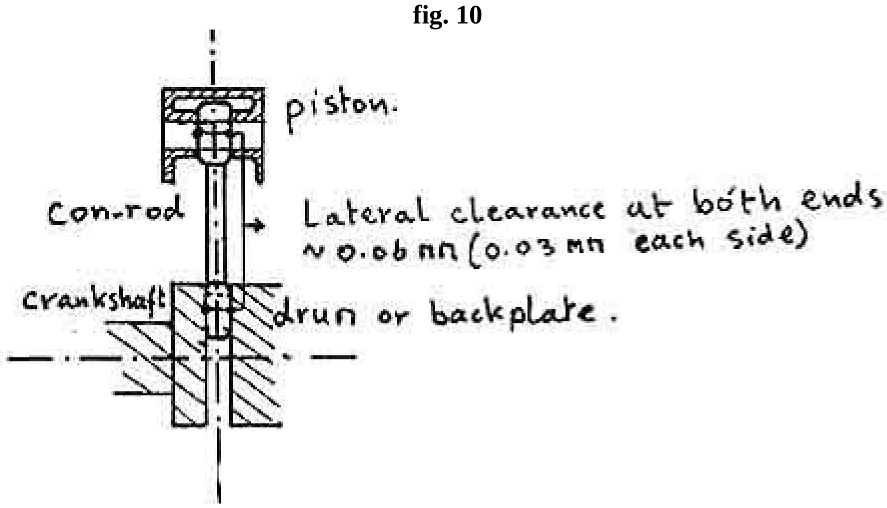
Our ideas on con-rod, piston cylinder assembly and cylinder head, as well as on the thermal aspects related to cooling will be descried here.

4.1 The con-rod.

Although being a simple part, the con-rod plays an important role in the engine. Since its weight is affecting the balance in the motor, it has to be made as light as possible.
Not having done experiments in thinning down con-rods to the point they will brake, we still use our original rod. It's main body has a cross section of 7 x 2,5 mm^2, the small end has a Æ4 x 4,5 journal bearing and the big-end one of Æ5 x 3 mm. Its weight is 1,9 grams.
Until now bronze bushes were used with a wall thickness of 0,25 mm.
There is absolutely no reason to use thicker bushes and it leaves more room for the dural (AISI 2024 type).
Our bushes were mounted with a 0.01 mm interference fit and loctite hot retaining compound.
Minimizing vibrations is an important part of our design philosophy so we gave a lot of attention to the free-play of the con-rod in all possible directions.
A play of ~ 0.025 mm on both crankpin and piston fit was found to be right and a few experiments on tighter and looser fits gave no dramatic effects. This is probably due to the fact that the difference in thermal expansion

between the steel pin and the con-rod is about ~ 0.01 mm at 150°C. The fit of the crankpin into the big-end shouldn't be too tight anyway, to allow the crankshaft bending during ignition.
In the description of the crankshaft positioning it was clearly stated that we want the con-rod as well as big-end exactly on the piston axis.
Limiting the con-rod lateral play on a big-end and small end location was the only solution we could find for the problem.
At both ends a lateral play of 0.06 mm has been used.
See also fig. 10.
Looking at con-rod small ends and piston pins told us, that in normal type piston the axial movement of the con-rod is considerable, introducing perpendicularity errors and increasing heat generation of both big- and small-ends.

fig. 10



Limiting lateral movement of the con-rod at both ends.

4.2 Piston and cylinder.

4.2.1 Choice of materials, lubrication.

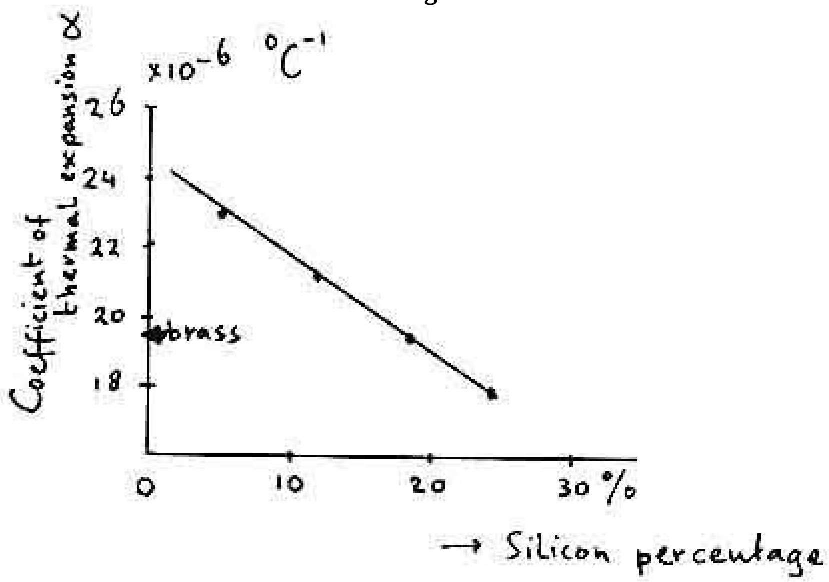
In order to minimize friction at higher temperatures it is necessary to make a piston of a material with a smaller expansion coefficient than the liner. The first experiments on Bugls, done by Enrico Flores and Visser-Buys, was to change from the original cylinder material (Phoenix- Triumphator with a coefficient of thermal expansion a of 10.0 up to 10.5. 10^-6 °C^-1) to a cylinder made out of a stainless steel type with a a of 14.5.10^-6 °C^-1. Supposing the piston and cylinder temperature to be equal, the piston fit for a piston out of Meehanite or similar: a = 10.0.10^-6 °C^-1 in the original cylinder remains about constant over a large temperature range. However, if the piston gets slightly hotter the fits gets tighter, increasing friction and thus heating up the piston more, eventually causing engine seizure.
The typical quick way Bugls MK I and MK II seize can probably be so explained.
With more expanding cylinder material friction reduces at higher temperatures, so the heat generation will be stabilized. This cylinder gave the Bugls a different character indeed! Even over-compressed or far too lean they would never seize, they only slowed down more or less, but went on.

The advantage of a "stable" system is that it can be set very near to the maximum performance, without risking an extra stop because of a sudden seizure.
In the worst case the penalty will be a few "hard" laps at the end of each tank. The old Bugls were to be set somewhat under maximum speed, not to risk a sudden seizure.
Strange enough some were better on this aspect (Louis Petersen!) and some were really critical.
This could probably be explained by slight variations in the a of different samples of materials of the same kind.

In Louis case the "lapping" of the piston to the only fit that would work in the air, still giving good starts, was probably of conclusive importance and .... quite a job as we can tell you!
Also the tapering of the liner has strong effects, but this will be discussed later.
In the FMV and the first production Nelsons Meehanite pistons were used in combination with very ordinary steel (free cutting steel) hard-chromium plated liners. Our type of free cutting steel is DIN 9 S Mn 28, an alloy with suitability for easy chromium plating. These steel liners have a a of about 11.5. 10^-6 °C^-1.
We'd like to find a steel, preferably free cutting (!) with a somewhat higher a:~12.5.10^-6 °C^-1, but didn't find it yet. In ABC and AAC the difference between a's can easily be chosen by variation of the silicon-percentage in the piston.

Fig. 11 shows the relation between a and silicon percentage in aluminium.
It shows also that an 18% Si-Aluminium alloy should be good for brass or a 12% Si-Aluminium alloy cylinder. The mentioned alloys can for instance be obtained in bars through dealers of Mahle-Germany. (Mahle 138 and 124 alloys respectively).

fig 11



Coefficient of thermal expansion a of aluminium (20-200 °C) as a function of silicon percentage.

Another point of interest is the choice of material combination for piston and cylinder with respect to friction. Friction is very strongly determined by lubrication. In the conditions we're talking about (high temperature, zero to a few XXXm space between the piston and cylinder) a viscous layer of oil cannot exist so a boundary layer type of lubrication is probable.
Heat generation as well as effectiveness of the lubrication is strongly affected by the surface properties.
Very little is known about this in a quantitative way, but people think, that porous materials help and cast iron on non hardened steel gives relatively little generated heat. Graphite in cast iron improves lubrication and wear characteristics. Cast iron on porous hard-chrome is better than aluminium on chrome. But with an 18% Si-Aluminium piston the chrome is in contact with AL-Si crystals and we know nothing about that.

Trying to find out about lubrication we were glad to hear from Emil Rumpel an indication of the cylinder wall temperature near the combustion chamber, measured in a model in a wind tunnel.
In a speed glow and a TR-diesel (both Rossi) temperatures were about 250°C.
Hearing this, we really started worrying, because oil vapors will burn spontaneously (flash) and stop lubricating at these temperatures range, impairing lubrication.

The following table shows the flash point of a few types of oil:

Oil type	**flash point**
Castor	275° - 290° C
Palgol (synt.)	220°C
Ucon 650 X (synt.)	253°C
ML 70 (synt.)	260°C

So Castor oil turns out to be the best also with an extended distillation range, so no sudden effects occur. Synthetic oils will suddenly burn after passing a certain temperature. This is one of the reasons why synthetic oils are so clean. They'll burn completely in and near the combustion chamber not leaving any sort of lacquer or carbon deposition. For all but the hottest glows synthetic oils can work very well, giving the advantage of a cleaner engine.

Another quite important point is the wear of piston and liner.
The use of chromium plated cylinders will minimize wear, giving also a reasonable low friction.
We are still convinced that a non-hardened steel liner is faster and more forgiving, but it is too much work to make a new one every contest.

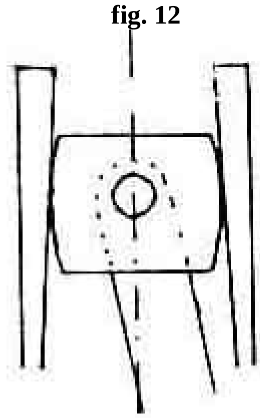
A few words on heat treatment of piston and liners...

Theoretically, stress-releasing of pistons and cylinders machined form bar stock is necessary to prevent permanent deformation after thermal and mechanical load occurring when running. But it appeared that even after many hours of real "hard" runs a cylinder turned from free turning steel without stress release treatment stayed round within 0.002 mm, equal to the initial value. Pistons from cast-iron have been tested in the FMV either or not stress-released and no significant effects in stability were found for Meehanite used by Bugl, Meehanite SFF40 and gray centrifugal cast iron. The performance of all these types of pistons was similar anyway. The only effect of the heat treatment of the piston (bringing it to 500-530°C and cooling it slowly), is that it "grows", so it won't "grow" during running anymore. This makes it easier to fit the piston to size and doesn't need running in to compensate for a growing piston, giving the same fit for a long time after a few minutes of running in. Not to be misleading we must add that stress-releasing treatments will probably be necessary in the case of a production engine, because machining according to production methods will introduce much more stress in the materials.

4.2.2. Shape of piston and cylinder.

In nowadays lapped piston 2,5 cc engines, tapered liners are widely used as well as a "barrel" shaped pistons (see fig. 12).

fig. 12



Tapered liner with barrel-piston

A taper liner gives better sealing at the point where pressure in the cylinder is high, and reduces friction where sealing is less important. From experiments on different engines we found a taper of the liner between 0.015 and 0.025 mm in diameter per cm. to give satisfying results. This is a bit less than Rossi, and about equal to Nelson and Bugl. A roundness of the liner within 0.0015 mm in diameter in the FMV cylinders turned out to be (too?) good. The only effect of ovality (up to 0.005 mm) seems to be more difficult starting, unless the piston is oval in the same way, but how can you make that!

The "barrel" shaped piston has two advantages:

Firstly a wedge is formed to press the oil (if any) between the piston skirt and the cylinder to improve lubrication.

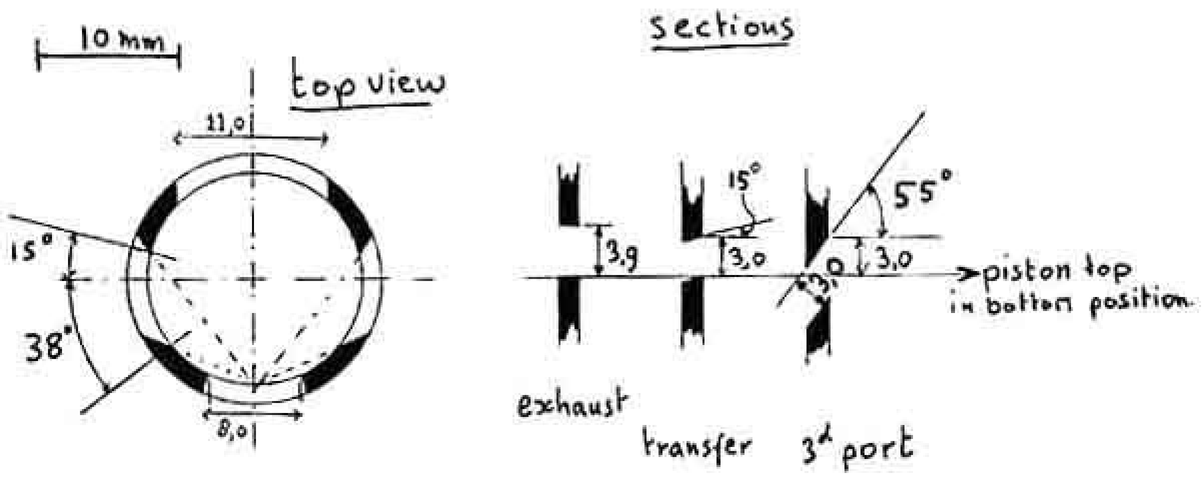
Secondly it allows the piston to "find its way" if the cylinder is deformed from thermal reasons.
The way we barreled the FMV's piston is drawn in fig. 14a. The angles were turned before pre-honing the pistons to about 0.005 mm from its size.

4.2.3 Transfer and exhaust ports in the liner.

As we have stated before, we don't believe in the effects of different portings.
We are convinced that the huge third port in the Nelson is the reason that it goes so well. Maybe it helps, but with a smaller third port is wouldn't be much worse. For a teamrace engine the exhaust width shouldn't be too wide, 10-11 mm is enough. A bigger opening there may waste fresh mixture. For mechanical reasons it is favorable to keep the ports as small as possible, giving more cylinder wall for the piston for guidance.
The inclination of the third port is in most engines 50°-60°, no differences were found between main transfer ports inclined between 0° and 15°.

In figure 13 a section of the cylinder parts is drawn.

fig.13



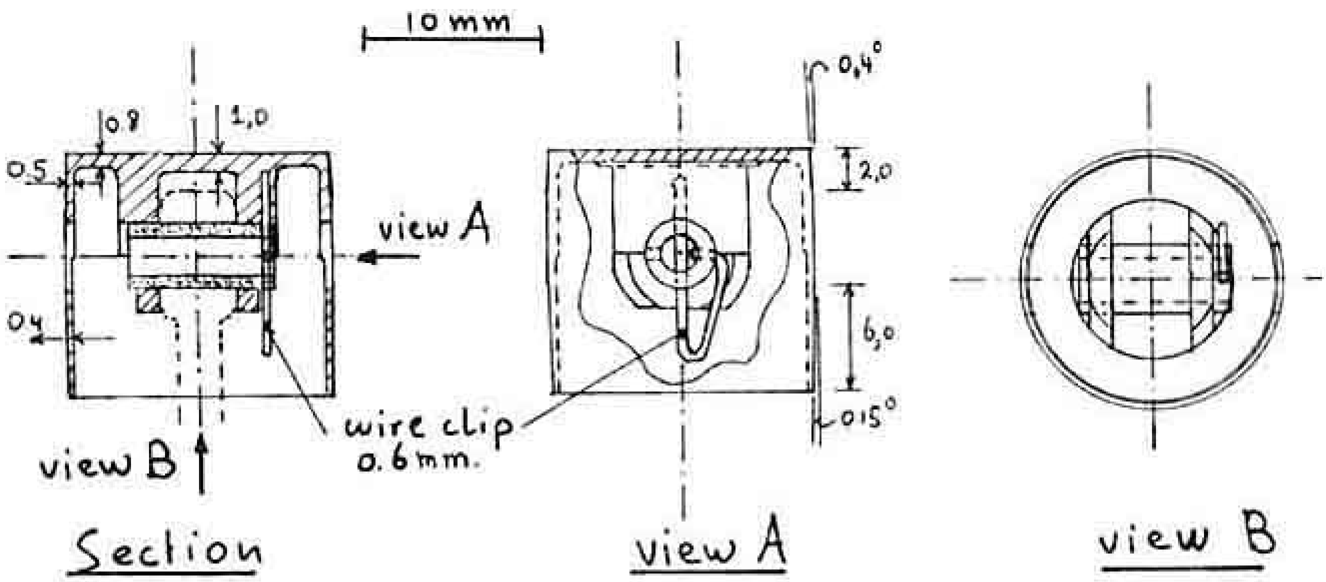
FMV cylinder ports

4.2.4 Piston design and cooling.

About the outside shapes of the piston a few things were said before.
In the following a few other aspects of the FMV piston design are treated.

The design had to meet two demands: lightness and limited con-rod end play.
With a piston pin directly supported by the top of the piston a very thin (0.4 - 0.5 mm) piston skirt is possible.
The small end construction is shown in fig. 14a.

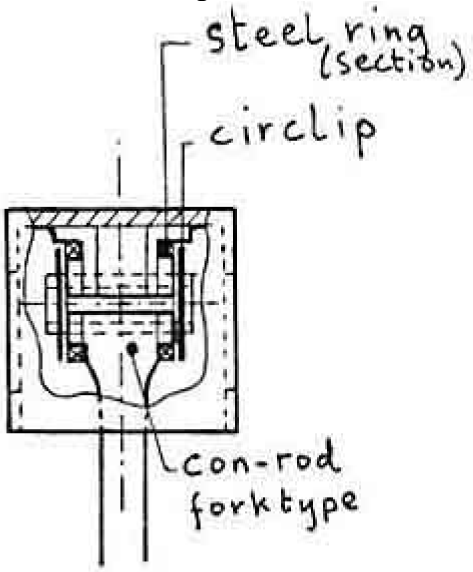
fig 14. a



FMV piston & piston pin

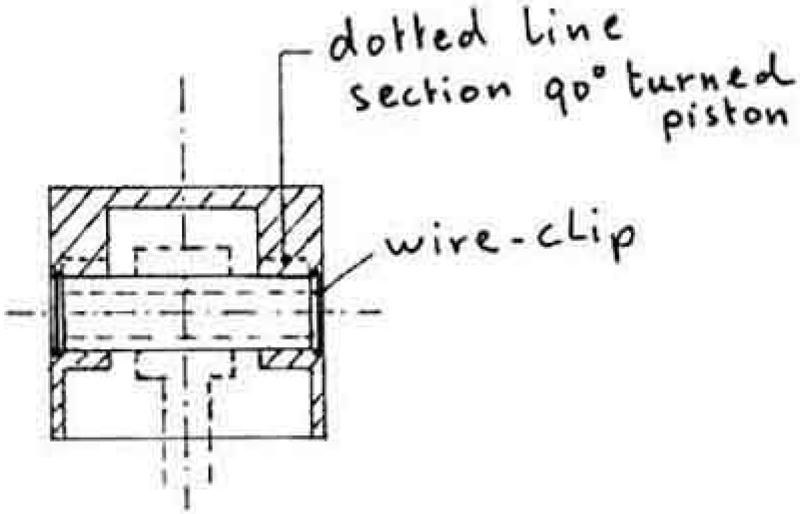
The FMV cast iron piston weight is around 3.6 gram, to compare: Rossi cast iron 5.2 gram; K&B-Nelson 6.1 gram. The piston pin is short, and thus light at 0.6 grams, 0.4 gram lighter than Rossi and Nelson. In principle Bugl's set up (fig. 14b) has the same advantages, but piston + piston-pin construction is still about 5.2 gram.

fig 14b



Bugl piston side view

fig. 14c



Nelson, K&B piston section

The piston pin in the FMV is kept in place by a wire clip of 0.6 mm diameter going through the pin into the piston, see fig. 14a. The thing looks nice and simple, but needs further improvement in order to enable the separation of the con-rod and the piston in the motor. With our small end construction it is quite a hard job to get the con-rod of the shaft with the piston on.

By chance we found out that a very thin piston skirt has one more advantage. Because the piston can be easily radially compressed the diameter of the piston "follows" the taper of the liner, making the motor almost completely insensitive to piston fit. It acts more or less like a piston ring. Unlike a Rossi-type piston, that will stick suddenly when pushed by hand towards T.D.C in a tapered liner, the FMV piston only gives a gradual increase in resistance. No fatigue failure in the piston was found until now, so maybe thinner walls are possible for improved performance.

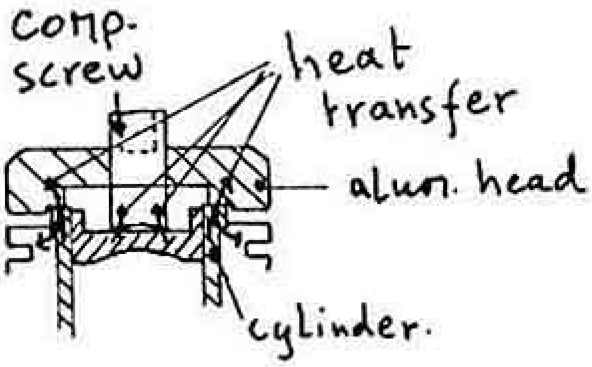
Since it was shown before that a too high piston wall temperature causes excessive expansion and lack of lubrication, the cooling of the piston is important. There are two main ways the piston can be cooled.
In first place there's cool air and fuel at the underside of the piston. If, like in the Bugl all fresh gases flow through the piston, this can be a main way of cooling. In Nelsons, Rossi's and FMV's we can only hope for "refreshment" from that side. In the second place, there is a given amount of metallic contact between piston and liner, especially with tight fits. With a loosely fitted piston, hot gas will leak between piston and liner heating up the walls and preventing the already limited metallic contact. This must be the main reason that worn out pistons and liners overheat quite easily.

One of the main sources of the heating of the piston will be radiation from the combustion process.
A black piston (carbon deposition) receives far more heat than a shining type. So the cleaner it is, the cooler it stays. Striving for higher thermal efficiency with inevitable increased gas temperature, the piston temperature could turn out to be a bottle neck and may be it already is, due to possible oil flashing. At this moment we're thinking about a super shining piston top as a useful way to limit piston temperatures.
Preventing carbon built up might be our future goal.

4.2.5 Cylinder head and cylinder cooling.

From the discussed upper limit in cylinder- and piston wall temperature improved efficiency calls for improved cooling. In speed engines a power increase will normally go together with a more than proportionally increasing fuel consumption to give more internal cooling. It is not by chance that in the last years more and more cylinder heads with improved heat transfer from the cylinder and combustion chamber to the outside were seen in team racing. The Russians, Larsson-Rylin, Nelson and last year Bugl too, changed from the traditional steel contra-piston, directly fitted in the cylinder (fig. 15a), to an all aluminium alloy head.

fig. 15a

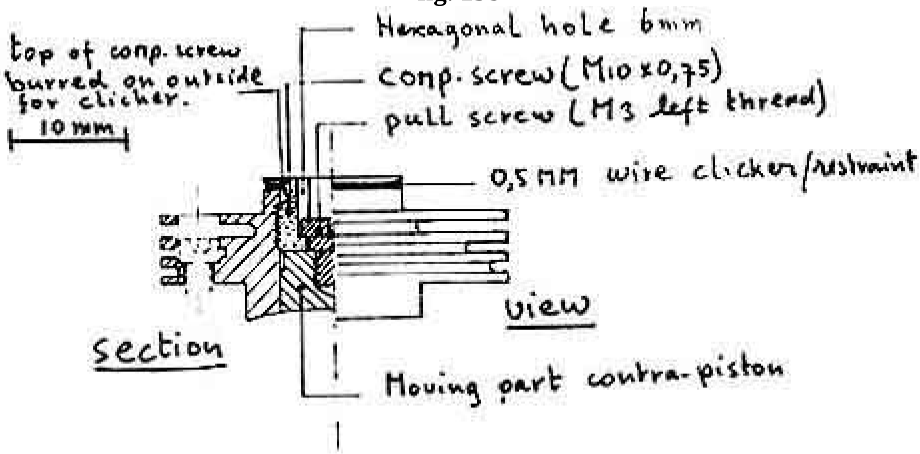


Traditional steel contra piston

In the old system the cylinder is hardly cooled because at rising temperature there is no direct contact between cylinder and crankcase due to the more expanding aluminium crankcase. There's no way to keep a metallic contact above 150°C (with a steel cylinder). It would need an interference fit at room temperature of 0.025 mm or more, which is quite impossible for different reasons. The all aluminium head with a small moving contra-piston, gives a good thermal contact between the cylinder and outside because the head insert expands more than the cylinder.

In fig. 15b the FMV version of this head is drawn. The compression is adjusted by a push-pull mechanism, avoiding the necessity of a very accurate tolerance on contra-piston fit.
An interference fit of about 0.005 mm is used with a moving contra-piston of 18% SI-aluminium alloy to counter seizure and to get a more constant fit over a wide temperature range.

fig. 15b



FMV cylinder head

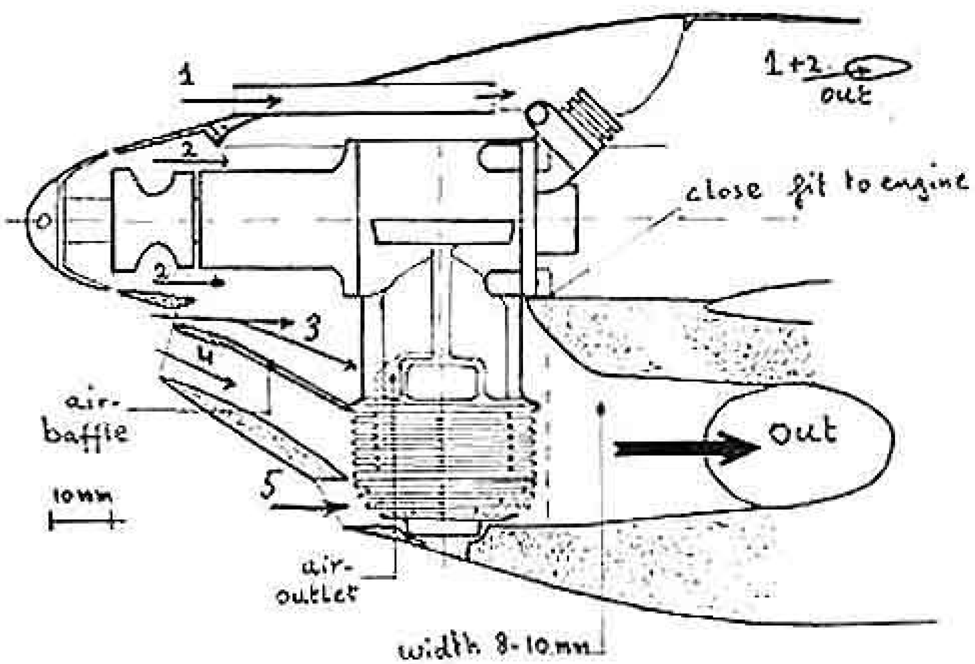
Since the conductivity of heat is better in aluminium than in steel, this may turn out to be a strong argument to change to aluminium cylinders. A better heat conductivity also means smaller internal temperatures differences,

also an advantage with respect to thermal deformation. A logical next step to a better cooled cylinder will possibly be a cylinder with integral cooling fins like HGK and the small Cox engines.
Thermal deformation of the cylinder wall due to temperature differences caused by cooling air entering from one side is the problem to be solved. We are sure, that when all the mechanical problems in the motor will be solved, only a better control of the piston and cylinder wall temperature will enable further improvements in efficiency.

5. The cooling of the motor in the model.

The goal of all cooling systems is to extract enough heat from the motor and limit thermal deformation. Temperature differences have therefore to be minimized. In the FMV model of '78 the cooling air supply was divided in 5 separate parts.
Fig. 16 shows the way the air was supplied (a bit different form Dave Clarkson's interpretation in Aeromodeller Nov. '78. It really was your own idea Dave, and probably not a bad one too!).

fig. 16



Cooling in the FMV Model
1. 4.5 mm . i.d. pipe to carburettor
2. air to crankcase bottom (blocked by spinner)
3. air to crankcase middle & exhaust
4. air to cylinder ins
5. 8 mm. i.d. to cylinder head
air-baffle is more or less adjustable

The 4.5 mm i.d. "pipe" directing to the carburetor is extremely important in our view. Air from a constant temperature is the only way to get constant needle settings. The only air of constant temperature is air directly from outside and this is also the coolest air we can get.
Vapour lock problems, being more common than most people like to think, are solved by this approach. In the FMV model the crankcase bottom (ballraces etc.) was not directly cooled for the earlier mentioned reasons. For an all aluminium crankcase cooling is absolutely necessary.
The crankcase between the cylinder fins and the bottom gets a moderate amount of air. Some of this air is directed to the exhaust and leaves the model there through a hole near the exhaust.
The air to the cylinder fins is pressed through by making the cooling duct very closely fitted to the motor.

The air is directed to the head by a tube and pressed through the fins in the same way as for the cylinder. All this cooling air openings are more or less independently variable by changing the tube diameters or the setting of the air baffle thus changing the distribution of the air between the cylinder fins and the exhaust/ middle crankcase part.

This way of a regulated cooling for each part of the motor will only work, if the main air resistances in the system are the ones we can vary. So for the rest of the cooling ducting an air resistance as low as possible is necessary. A relatively large air outlet is therefore used.

The problem of temperature difference between the front and the rear of the cylinder is not solved in our system. Until now it doesn't seem to be important (for a side exhaust engine like the FMV at least!) because the thermal deformation of the crankcase (bending and getting an egg-like section) doesn't "touch" the steel cylinder at running temperatures. With an aluminium cylinder and/ or integral cooling fins, this problem may become more important.

-oOo-