HOW THE NUMBER OF ENGINE CYLINDERS AFFECTS ITS MECHANICAL EFFICIENCY

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Abstract

The relation between engine principal dimensions, i.e. piston diameter to stroke ratio is the parameter often modified in order to increase engine efficiency and durability. A similar operation is the achievement of given volume for different number of cylinders, which is the subject of the following paper.

For recent years the engine manufacturers have been proposing in-line engines of 3, 4, 5 and 6 cylinders more eagerly than in last century. Obviously engines of lesser cylinder number are proposed for small cars while those of greater number – for most dynamic ones. One can ask a question whether the cylinder number for a certain swept volume affects such parameters as engine efficiency and durability as well as lube oil consumption. There is no simple answer to this question but simulations performed by the author allow estimating tendencies in changes of those parameters for various numbers of cylinders. Moreover, it is quite vital that engine complex modifications accompanying the change in cylinder number have been taken into consideration in a course of carried out tests. Lesser number of cylinders for the same volume results in more solid pistons and connecting rods, worse engine run irregularity and so on.

The paper presents introductory conclusions which allow selecting an engine of the cylinder number that ensures the best economical and dynamical parameters for a given car.

Keywords: *mechanical efficiency, oil consumption, irregularity run*

1. Dependence of Engine Constructional Features on its Swept Volume

Implementation of an assumed swept volume to lesser cylinder number requires a design of crank mechanism elements of relatively increased dimensions. When analyzing dimensional relations of currently produced elements of crank mechanism one can notice certain regularities which do not deny numerous deviations from this rule. It can be assumed with high level of approximation:

- masses and volume of elements are proportional to single cylinder swept volume,
- linear dimensions vary proportionally to the cube root of swept volume change,
- cross-section area varies proportionally to the square of volume change cube root.

Taking these regularities into consideration a procedure written in PASCAL (presented in Tab. 1) has been formulated [1, 2].

4-cylinder engine of 76.5 mm cylinder diameter has been assumed as the standard one. For example, changing the cylinder number to 5 the cylinder diameter should be multiplied by the factor of *LL=L5* equal to 0.9383 in order to keep the swept volume unchanged. Eventually, the rank radius named *korba* in the procedure 1 should be multiplied by the same factor *LL* in order to keep unchanged the stroke to diameter ratio S/D. Monitoring entries in procedure 1 one might conclude how the change in cylinder number affects other parameters of engine design.

While keeping the constant value of S to D ratio does not raise controversies, keeping proportions is not so obvious. As to just roughly justify assumption of constant piston height to diameter ratio the Fig. 1 presents two today produced pistons that in author's opinion are representative for modern motor industry.

Tab. 1. Dimensions and masses for defined number of cylinders

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{DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE}
      mno =1.00;                    {zwiek.  srednicy tloka dla stal.pojemn }
      L1=1.5874;                    {zmiana liczby cylindrow dla stal.pojemn}
      L2=1.2600;
      L3=1.1006;
      L4=1.0000;
      L5=0.9383;
      L6=0.8738;
      L8=0.7937;
      lcyl=   4;                    {liczba cylindrów w silniku rzędowym    }
      LL  =  L4;
      korba   =0.0756/2/mno/mno*LL; {promień korby                          }
      korbowod=0.144   /mno/mno*LL; {długość korbowodu                      }
      d       =0.0765  *mno    *LL; {średnica cylindra                      }
      mo      =(0.26+0.081+0.080)*mno*mno*mno*LL*LL*LL; {masa oscyl         }
      mwir    = 1.1*LL*LL*LL;       {masa obrotowa                          }
      ra=korba           ;
      lam=ra/korbowod     ;         {LAMBDA                                 }
      du =d               ;         {srednica cylindra                      }
      kolr=0.030 *mno *LL ;         {odległ. piersc. od dolnej kraw. tłoka  }
      kolb=0.020 *mno *LL ;         {odlegl. sworznia od kraw. tloka        }
      multp3= 1.0e-7;
      exzentp3=-20;
      multp2= 1.0e-6 ;exzentp2= 0 ; {okreslenie ksztaltu pier. dolnego      }
{6e-7}multp1= 5.0e-7 ;exzentp1= 0 ; {okreslenie ksztaltu pier. gornego      }
      multko=15.0e-6 ;exzentko= 10; {okreslenie ksztaltu pow. bocznej tloka }
      zrodlo='oblicz';              {'trans?'- pobiera dane z katal. c:\film}
                                    {utworz. w programie c:\kolben\tlok.pas }
                                    {'oblicz'- na podst. multko i exzentko  }
                                    {'reczn?'- ksztalt wg. recznie tworzone.}
                                    {pliku o nazwie c:\film\reczn?.out      }
                                    {fiatkolb daje mozliwosc dwubarylkowatos}
                                    {ksztalt tloka pobierany jest w progr.  }
                                    {fiatkolb.pas ze zb. trans?.out         }
      hr1=0.00120*LL*mno   ;
      hr2=0.0012 *LL*mno   ;        {okreslenie wysokosci pier. gorn. i dol.}
      hr3=0.00040*LL*mno   ;
      ht4=0.033  *LL*mno   ;        {okreslenie wysokosci pier. zgar. i tło.}
      ps1= 2.00e5   ;ps2=2.00e5;    {okreslenie nacisku   pier. gorn. i dol.}
      ps3= 1.00e6{6};ps4=1e3   ;    {okreslenie nacisku   pier. zgar. i tło.}
      a1=0.00570e-0*LL*mno ;
      a2=0.00480e-0*LL*mno ;        {odlegl. pier. gorn. i dol. oraz d. i z.}
      a0=1e-0      *LL*mno ;
      a3=0.017e-0  *LL*mno        ; {a0*(ht4/2+ht4/50*exzentko)             }
      GGG=10000000                ; {sprezystosc oleju smarujacego          }
                                    {50o-42.6  70o-22.7  90o-13.9 [mPas]    }
      ep=10.000     ;               {stopien sprezania                      }
      ro=1.0001     ;bet=0.8      ; {stop. przyrostu objetosci i relacja cis}
      n1=1.35       ;n2=1.38      ; {wykl. politropy spr. i rozpr.          }
      ms=0.0001     ;{0.0017}       {kat pochylen poprzecznych tlo. w [rad] }
      amp= - 0.0e-6 ;               {ampl. odkszt. cyl. dla "+" hiperboloida}
          {-50}                     {dla "-" barylka                        }
      tokres=0.160 ;                {okres odksztalcenia cylindra           }
      wverl=0.0{10+0.08};           {przemieszczenie wezla odkszt. cylindra }
                                    {w stosunku do GMP gornego pierscienia  }
      p0= 1.00e5   ;                {cisnienie otoczenia                    }
      pd=1.90e5    ;pw= 2.10e5   ;  {cisnienie ladowania i wylotu           }
      eta=0.0127                 ;  {lepkosc dynamiczna oleju w [Pas]       }
      om=round(5000*2*pi/60)     ;  {prędkość kątowa wału                   }
      fic=3.62                   ;  {stopień przyrostu ciśnienia            }
      z1=35.0e-6                 ;  {początkowa grubośćfilmu olejowego      }
      zwp= 14.0e-6               ;  {grubość warstwy oleju na gładzi cylin. }
{DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE DANE}
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Fig. 1. Comparison of dimensional relation between pistons of modern gasoline engines: piston of 86.4 mm diameter on the left hand side and piston of 76.5 mm diameter on the right hand side

The carried out measurements confirm the assumption that keeping certain relations of piston dimensions is independent on manufacturer. For this certain case the height H to diameter D ratio is 1.46 for bigger piston and 1.59 for the smaller one. Despite noticeable differences in piston crown shape (see Fig. 2) also the location of gravity centres is similar.

Fig. 2. Comparison of piston crown dimensional relation between pistons of modern gasoline engines: piston of 86.4 mm diameter on the left hand side and piston of 76.5 mm diameter on the right hand side

More information on dimensions and masses of various elements of crank mechanism, pistons in particular, one can find in literature [3] while the summary of several comparable pistons has been presented in Fig. 3.

Fig. 3. Comparison of pistons and connecting rods of modern gasoline engines

2. Simulation Results of Principle Operational Parameters for 1, 2, 3, 4, 5, 6 and 8-Cylinder in-line Engines of 1400 ccm Swept Volume

Table 2 presents:
- friction power of piston set
- friction power of all ring sets
- predicted specific oil consumption and
- engine run irregularity level

Tab. 2. Record of computer simulation parameters characteristic for operation of 1.4 litre volume engines of various cylinder number

Number of cylinders of in-line engine	Friction power of single piston set	Total friction power of piston rings	Lube oil specific consumption	Engine run irregularity level
	[W]	[W]	[g/kWh]	[-]
1	830	1407	0.116	1/21
2	804	1310	0.144	1/49
3	789	1257	0.166	1/89
4	784	1220	0.185	1/386
5	805	1240	0.199	1/206
6	780	1176	0.314	1/486
8	776	1144	0.366	1/652

Calculations of lube oil specific consumption and friction power of piston set and rings have been carried out according to methods and programs completely described in literature [1, 2]. The calculation set is supplemented with courses of:
- oil film thickness,
- thickness of oil layer left by consecutive rings and piston skirt,
- rings and piston friction force against cylinder bore,
- level of oil cover over piston skirt.

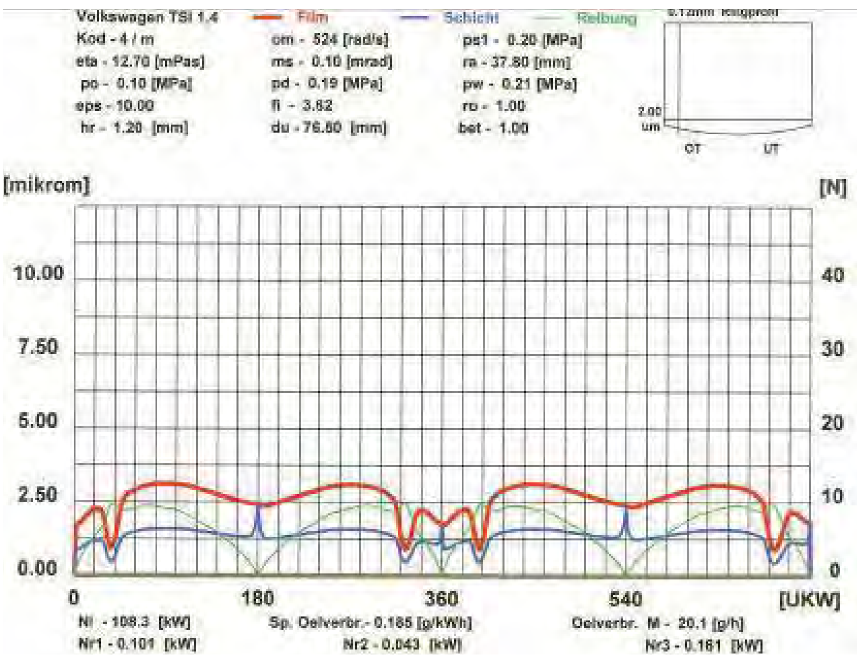

Fig. 4. Course of oil film thickness between upper compression ring and cylinder surface for a 4-cylinder engine – thickest line, course of oil layer thickness left over cylinder surface – medium size line and course of friction force of this ring

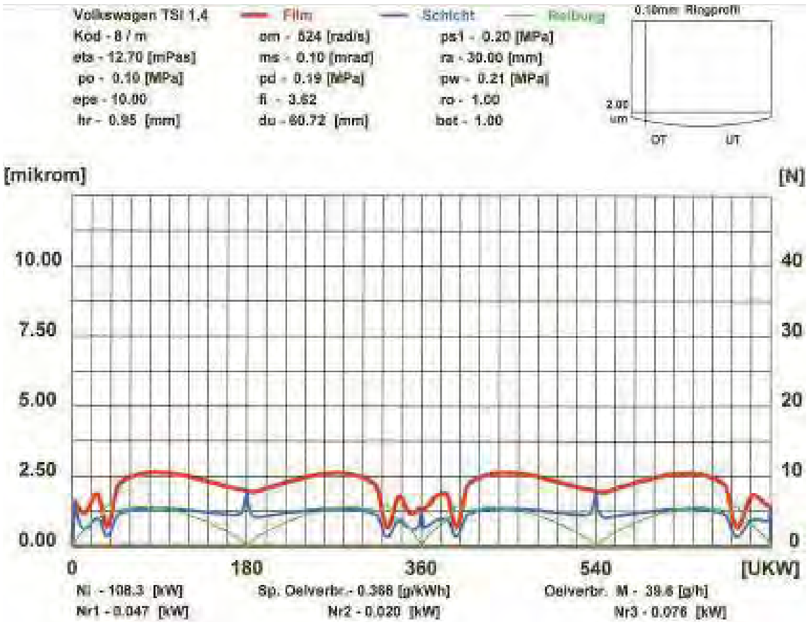

Fig. 5. Course of oil film thickness between upper compression ring and cylinder surface for an 8-cylinder engine – thickest line, course of oil layer thickness left over cylinder surface – medium size line and course of friction force of this ring for a single cylinder engine of volume as that in Fig. 4

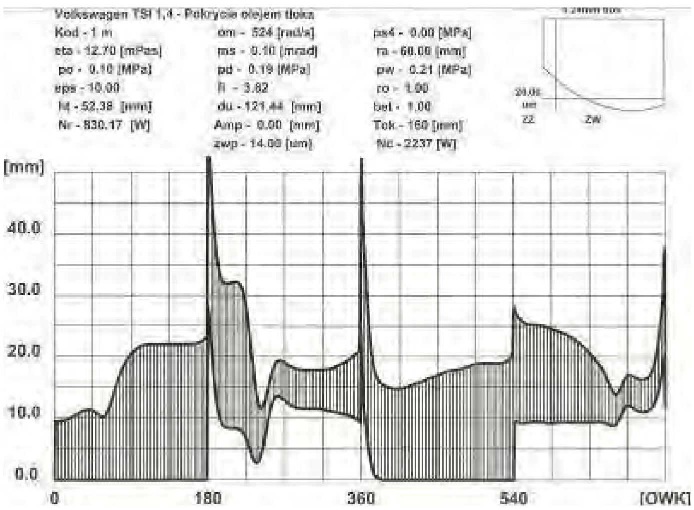

Fig. 6. Course of piston skirt oil cover for a single cylinder engine

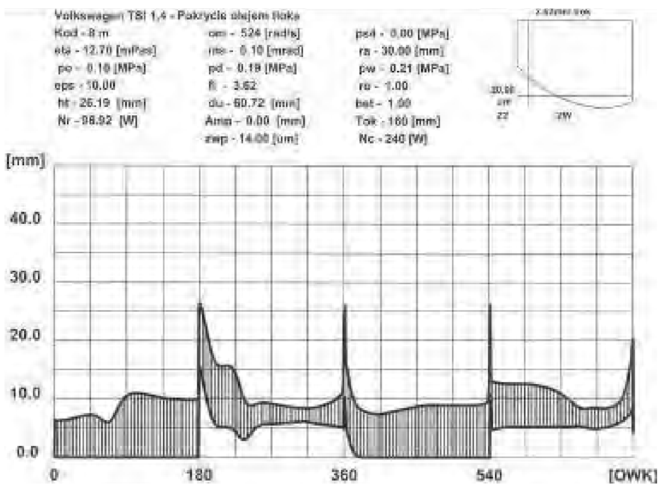

Fig. 7. Course of piston skirt oil cover for a single cylinder engine

Oil film cover over piston skirt presented in Fig. 6 and 7 illustrate the characteristic phenomenon of relatively poorer oil cover on skirt of smaller pistons. Consequently – despite the general belief – engines of higher cylinder number and unchanged swept volume do not demonstrate higher friction losses. The reduction in total friction losses on multicylinder engines gives an improvement of mechanical efficiency for such category of engines.

3. Analysis of Simulation Results

The results presented in previous chapter prove the advantageous effect of higher cylinder number on reduction in friction losses. This remark concerns not only pistons but also the ring packs. Even though the 5-cylinder engine disrupts this rule to minor extent, substitution a 1-cylinder engine with the one of 8 cylinders results in piston friction reduction by 7.7% while in case of all rings this gain amounts to 18.7%.

The advantageous effect of higher cylinder number on friction losses and eventually on mechanical efficiency does not mean that all parameters of engine operation would be improved. Unfortunately, higher number of cylinders equals higher oil consumption. For examples presented in Tab. 2 the oil consumption is over tripled for extreme cases.

The performed analysis of engine run irregularity gives easily predictable results, i.e. higher number of cylinders means higher engine run regularity. From this point, the 4-cylinder engine reveals interesting features. On that engine gas forces and forces of inertia are associated beneficially, because on long part of stroke they are directed reversely and partly reducing each other.

Summarizing, it should be noted that contrary to the general belief engines of higher number of cylinders and unchanged swept volume demonstrate better mechanical efficiency. Of course, there are engines of smaller cylinder number, e.g. 3-cylinder which consume less fuel than 4-cylinder engines of the same swept volume. However, in this case the reduction in specific fuel consumption results from other measures like more effective combustion in bigger volume of single cylinder.

4. Conclusions

1. Friction losses due to both pistons and piston rings can be reduced by the increase in cylinder number and unchanged total swept volume.
2. An increase in cylinder number for unchanged total swept volume brings about a considerable increase of oil consumption.
3. Engines of even cylinder number present more regular run in a case when forces of inertia are comparable to gaseous forces.

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