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Metode za oblikovanje elementov sesalnega zbiralnika batnega motorja z notranjim zgorevanjem

Design Methods for the Intake-Manifold Elements of Reciprocating Internal Combustion Engines

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V uvodu prispevka so prikazane metode oblikovanja sesalnih zbiralnikov, ki omogočajo povečanje prostorninskega izkoristka zaradi dinamičnih sprememb tlaka. Predstavljene so analitične metode za optimiranje premera in dolžine sesalnih cevi glede na vrtilno frekvenco motorja ter metode, ki upoštevajo resonanco v sesalnem zbiralniku. V osrednjem delu prispevka je predstavljena analiza učinkovitosti posameznih metod, ki je bila izvedena s simulacijo na modelu štirivaljnega motorja z notranjim zgorevanjem. Pri simulaciji so bili upoštevani fizikalni in kemični učinki od trenutka, ko pride zrak v sesalni zbiralnik, do trenutka, ko izpušni plini zapustijo izpušni zbiralnik. V prispevku je prikazan tudi način uporabe enorazsežnega modela za analizo vpliva izbranega elementa sesalnega sistema na prostorninski izkoristek motorja ter za izbiro optimalne rešitve pri danih zahtevah.

(Ključne besede: motorji z notranjim zgorevanjem, zbiralniki sesalni, optimiranje, simuliranje, metode analitične)

Methods for intake-manifold design that will lead to an increase in volumetric efficiency by using dynamic changes of pressure, are presented in the introductory part of this paper. Analytical methods for tuning the intake-pipe length and diameter to a specific engine speed, and methods dealing with the resonance in the intake manifold are considered. The main part of the paper comprises an analysis of these methods that was conducted on a simulation model of a four-cylinder spark-ignition engine. Physical and chemical processes are considered in the model, from the moment the air enters the intake system until the combustion gases leave the exhaust pipe. It is also shown how one-dimensional simulation calculations can be used for the analysis of a single intake-system element's influence on the volumetric efficiency of the engine, and for the selection of the optimal solution for given demands.

(Keywords: internal combustion engines, intake manifold, optimization, simulation, analytical methods)

0 UVOD

Na tok plina skozi motor z notranjim zgorevanjem v glavnem vplivajo sesalni in izpušni sistem ter konstrukcija in krmiljenje ventilov. Lahko torej rečemo, da je sesalni sistem zelo pomemben del motorja, katerega oblika in izmere vplivajo na prostorninski izkoristek motorja, porabo goriva in hrup. Med delovanjem motorja prihaja v sesalnem sistemu do dinamičnih sprememb, kar vodi do spremembe njegove učinkovitosti v odvisnosti od vrtilne frekvence motorja. S spremembo geometrijske oblike sesalnega sistema je pri dani vrtilni frekvenci motorja mogoče povečati njegov prostorninski izkoristek. Sesalni sistemi tekmovalnih motorjev so prilagojeni doseganju največjega prostorninskega

0 INTRODUCTION

Fluid flow through an internal combustion engine is mainly influenced by the intake system, the exhaust system and by the valve mechanism. Therefore, the intake system is an important engine element whose shape and dimensions influence the volumetric efficiency, fuel consumption, and noise pollution. During the engine's operation, dynamic changes occur in the intake manifold, which leads to a change in its efficiency with the change of the engine's speed. With a change to the geometry of the intake system, it is possible to increase the engine's volumetric efficiency at a specific engine speed. The intake systems of racing engines are tuned to produce maximum volumetric efficiency at a high engine izkoristka pri visokih vrtilnih frekvencah motorja, pri večini preostalih motorjev pa doseže sesalni sistem največji prostorninski izkoristek pri nižjih vrtilnih frekvencah motorja. Za zmanjšanje vpliva geometrijske oblike sesalnega zbiralnika na obnašanje motorja so bili razviti sesalni zbiralniki s spremenljivko geometrijsko obliko. S spreminjanjem dolžine sesalnih cevi dosežemo povečanje prostorninskega izkoristka motorja v širokem območju vrtilnih frekvenc. Za določitev izmer takšnega sesalnega sistema pa moramo poznati vpliv geometrijske oblike na prostorninski izkoristek in seveda tudi same metode za izračun izmer sesalnega sistema.

speed, whereas with most other engines, the intake systems produce maximum volumetric efficiency at a lower engine speed. To overcome different intake-manifold geometry effects on the engine behaviour, variable intake manifolds are being developed. The increase in the volumetric efficiency over a broad range of engine speeds is achieved by varying the intake runner. In order to determine the dimensions of such a manifold, it is necessary to be familiar with the influence of the intake-manifold geometry on the volumetric efficiency, as well as with some methods for calculating the manifold dimensions.

1 IZRAČUN GEOMETRIJSKE OBLIKE SESALNEGA ZBIRALNIKA

Za izračun geometrijske oblike sesalnih zbiralnikov je bilo razvitih več metod, ki jih lahko razdelimo tri razdelimo v tri skupine:

1. **Analitične metode** za izračun optimalne vrtilne frekvence pri danih izmerah sesalnega zbiralnika.
2. **Enorazsežne numerične simulacije** za izračun izmenjane količine plina med delovanjem motorja.
3. **Trirazsežne numerične simulacije** za izračun izmenjane količine plina med delovanjem motorja.

Metode, ki spadajo v tretjo skupino, so časovno zelo potratne, zato niso primerne za analizo celotnega sesalnega zbiralnika, temveč le za posamezne manjše dele, npr. za simulacijo toka plina skozi sesalni ventil. Zaradi tega metode v tretji skupini ne bodo opisane bolj podrobno.

1 INTAKE-MANIFOLD GEOMETRY CALCULATIONS

In the past a large number of expressions and methods for calculating the dimensions of intake manifolds have been developed. They can be divided into three main groups:

1. **Analytical expressions** that calculate the tuning engine speed using manifold dimensions.
2. **One-dimensional simulation calculations**, which calculate the amount of fluid that is exchanged during the engine's operation.
3. **Three-dimensional simulation calculations**, which calculate the amount of fluid that is exchanged during the engine's operation.

This third group of methods is not suitable for the entire intake manifold because these methods consume a considerable amount of time for the model design and calculation. But they are useful for the simulation of individual small parts, such as the flow through the intake valve. Therefore, this group of methods will not be described in more detail.

1.1 Analitične metode

Sesalne zbiralnike, ki vodijo do povečanja prostorninskega izkoristka motorja, lahko razdelimo v dve skupini: v *optimirane* in *resonančne* sesalne zbiralnike.

Optimirani sesalni zbiralniki dosežejo tlačno konico pri določeni vrtilni frekvenci motorja kar vodi do največjega prostorninskega izkoristka. Raziskave so pokazale, da je prostorninski izkoristek največji v primeru, če doseže tlak v sesalnem zbiralniku največjo vrednost v območju zasuka glavne gredi 20 do 50 stopinj pred zaprtjem sesalnega ventila. Vrtilna frekvenca motorja, pri kateri pride do največje vrednosti, imenujemo optimalna vrtilna frekvenca.

Po [1] lahko na podlagi predpostavke o poteku tlakov v sesalnem zbiralniku pred sesalnim ventilom določimo nihajni čas, ko so sesalni ventili odprti t_1, oziroma teče plin v valj, in čas, ko so ventili zaprti t_2, oziroma ni pretoka. Na podlagi teh dveh nihajnih časov izračunano dolžino l_p in prerez A_p

1.1 Analytical expressions

The intake manifolds that result in an increase of engine's volumetric efficiency can be divided into *tuned intake manifolds* and *resonant intake manifolds.*

At some engine speed **a tuned intake manifold** causes a pressure trace in the manifold that leads to the maximum volumetric efficiency. Research has shown that if the pressure in the intake manifold is a maximum in the period of 20–50 crank angle degrees (CA deg) before the intake valve closes, then the maximum volumetric efficiency is obtained. The engine speed at which this maximum occurs is called the tuned engine speed.

According to [1], from the assumed pressure trace in the intake manifold in front of the intake valve, the oscillation time period when the valves are open t_1, and when the valves are closed t_2 are determined. With these time periods, the length l_p, and a cross-sectional area A_p of the primary intake pipe are glavne sesalne cevi po enačbah (1) in (2):

calculated using Eq. (1) and (2):

$$l_{\mathrm{p}} = c \cdot \frac{720° - \alpha_{\mathrm{ivo}}}{24 \cdot \kappa_{\mathrm{c}} \cdot n} \quad , \quad \mathrm{m} \tag{1}$$

$$A_{\mathrm{p}} = \frac{12 \cdot \pi \cdot \kappa_{\mathrm{o}} \cdot V_{\mathrm{c}} \cdot n}{c \cdot \alpha_{\mathrm{ivo}}} \cdot \tan\left[90 \cdot \kappa_{\mathrm{o}} \cdot \frac{720 - \alpha_{\mathrm{ivo}}}{\kappa_{\mathrm{c}} \cdot \alpha_{\mathrm{ivo}}}\right] \quad , \quad \mathrm{m}^2 \tag{2},$$

kjer so: c v m/s – hitrost zvoka; n v min^{-1} – vrtilna frekvenca motorja; $t_{\mathrm{ivo}}=\alpha_{\mathrm{ivo}}/(6{\cdot}n)$ in $t_{\mathrm{ivc}}=\alpha_{\mathrm{ivc}}/(6{\cdot}n)$, s – čas ko je sesalni ventil odprt oz. zaprt.; V_{c} v m^3 – prostornina valja; α_{ivo} in α_{ivc}, ° - zasuk glavne gredi, ko je sesalni ventil odprt oz. zaprt; $\kappa_{\mathrm{o}}=t_{\mathrm{ivo}}/t_1$ in $\kappa_{\mathrm{c}}=t_{\mathrm{ivc}}/t_2$ – razmerje med časom odprtja/zaprtja sesalnega ventila in pripadajočim nihajnim časom.

Posledica spreminjanja tlaka v sesalni cevi v času, ko so sesalni ventili odprti, so tlačna nihanja, ki se ohranijo tudi po trenutku, ko se sesalni ventil zapre. Po [2] lahko ta preostala tlačna nihanja v sesalni cevi še dodatno povečajo prostorninski izkoristek, če se največji tlačni vrh preostalega nihanja ujame z zgornjo mrtvo lego (ZML) pri sesalnem taktu. Enačba za popis omenjenega stanja je naslednja:

where: c, m/s – speed of sound; n, rpm – engine speed; $t_{\mathrm{ivo}}=\alpha_{\mathrm{ivo}}/(6{\cdot}n)$ i $t_{\mathrm{ivc}}=\alpha_{\mathrm{ivc}}/(6{\cdot}n)$, s – time during which the intake valve is open or closed; V_{c} , m^3 – cylinder volume; α_{ivo} i α_{ivc}, deg - crankshaft angle during which the intake valve is open or closed; $\kappa_{\mathrm{o}}=t_{\mathrm{ivo}}/t_1$ i $\kappa_{\mathrm{c}}=t_{\mathrm{ivc}}/t_2$ –ratio of the valve timings with oscillation time periods.

As a result of pressure changes in the intake pipe during the time the intake valves are open, the pressure in the pipe continues to oscillate after the intake valve closes, and these oscillations are called residual waves. According to [2], if a peak pressure of the residual wave from the previous cycle occurs at the top dead centre (TDC) of the intake stroke, an additional improvement in the volumetric efficiency is obtained. From this approach, the following equation is derived:

$$\left(2 \cdot k - 1\right) \cdot \theta_{\mathrm{t}} + \theta_{\mathrm{d}} = 720 \tag{3},$$

kjer je: $\theta_{\mathrm{t}}=(12{\cdot}n{\cdot}l_{\mathrm{p}})/c$ v stopinjah – zasuk glavne gredi za čas potovanja tlačnega vala od valja do sesalne cevi in nazaj; θ_{d} v stopinjah – zasuk glavne gredi za čas sesalnega pulza v sesalni cevi.

Po [4] izračunamo optimalno vrtilno frekvenco motorja kot:

where: $\theta_{\mathrm{t}}=(12{\cdot}n{\cdot}l_{\mathrm{p}})/c$, CA deg – time period of wave travel from the engine cylinder to the pipe end and back; θ_{d}, CA deg – time period of a suction pulse in the intake pipe.

Ref. [4] proposed a slightly modified, simpler expression for calculating the tuned engine speed:

$$n = \frac{1}{q_{\mathrm{s}}} \cdot \frac{\theta^*}{24} \cdot \frac{c}{l_{\mathrm{p}}} \quad , \quad \mathrm{min}^{-1}/\mathrm{rpm} \tag{4},$$

kjer so: q_{s} - število celih tlačnih valov v preostalem tlačnem nihanju; $\theta^*=540+\theta_{\mathrm{ivc}}$ v stopinjah – zasuk glavne gredi, v katerem ta preostala tlačna nihanja obstajajo; θ_{ivc} v stopinjah – zasuk glavne gredi pri zaprtem sesalnem ventilu po spodnji mrtvi legi (SML).

Pri **resonančnem sesalnem zbiralniku** vpliva na povečanje prostorninskega izkoristka motorja ujemanje tlačnih nihanj z resonančno frekvenco sesalnega zbiralnika ali enega izmed njegovih delov. Največji prostorninski izkoristek ponovno dosežemo le pri določeni vrtilni frekvenci motorja.

V viru [4] so tlačna nihanja v sesalnem zbiralniku razdeljena na dva osnovna tlačna vala. Prvi tlačni val je posledica oblike glavne sesalne cevi in ima kratko periodo, medtem ko na drugi tlačni val z daljšo periodo vpliva oblika celotnega sesalnega zbiralnika. Tam so bili izpeljani tudi izrazi za izračun resonančne frekvence ω (rad/s) sesalnega zbiralnika s štirimi posamičnimi glavnimi sesalnimi cevmi (sl. 1):

where: q_{s} - the number of complete residual wave oscillations; $\theta^*=540+\theta_{\mathrm{ivc}}$, CA deg – period in which residual waves exist; θ_{ivc}, CA deg – angle of intake valve closure after BDC.

At a defined engine speed **the resonant intake manifold** causes an increase in the volumetric efficiency due to the correspondence of the pressure disturbance frequency with the intake manifold resonant frequency, or one of its parts.

Ref. [4] concluded that pressure oscillations in the intake manifold are comprised of two basic waves. One wave, which is influenced by the shape of the primary intake pipe, has a short oscillation period, and the other, which has a longer oscillation period, is influenced by the whole intake manifold. It also derived expressions for calculating the resonant frequency ω (rad/s) of the intake manifold with four individual primary intake pipes (Fig.1):

$$\cos\frac{\omega \cdot l_{\mathrm{p}}}{c} = 0 \tag{5}$$

$$\frac{A_{\mathrm{s}}}{A_{\mathrm{p}}} \cos\frac{\omega \cdot l_{\mathrm{s}}}{c} = \frac{\omega \cdot V_{Sb}}{A_{\mathrm{p}} \cdot c} + 4\tan\frac{\omega \cdot l_{\mathrm{p}}}{c} \tag{6},$$

kjer so: A_s v m^2 – prerez vstopne sesalne cevi; l_s v m – dolžina vstopne sesalne cevi (to je cev, ki je pred zbirnim prostorom sesalnega zbiralnika); V_{Sb} v m^3 – prostornina zbirnega prostora sesalnega zbiralnika.

Vir [5] pa je dopolnil zgornja izraza še z upoštevanjem vpliva prostornine valja. Enačba (7) podaja resonančno frekvenco za glavno sesalno cev, enačba (8) pa resonančno frekvenco za celoten sesalni zbiralnik:

where: A_s, m^2 – cross-sectional area of the secondary intake pipe; l_s, m – length of the secondary intake pipe (the pipe that is located before the manifold plenum); V_{Sb}, m^3 – intake manifold plenum volume.

Ref. [5] considered the influence of the cylinder volume. With Eq.(7), the resonant frequency ω that is related to the primary intake pipe is calculated, whereas by Eq.(8) the resonant frequency related to whole intake manifold is calculated:

$$\frac{A_c}{A_p}\tan\frac{\omega\cdot l_c}{c}\tan\frac{\omega\cdot l_p}{c}=1 \qquad (7)$$

$$\frac{A_s}{A_p}\cot\frac{\omega\cdot l_s}{c}=\frac{\omega\cdot V_{sb}}{A_p\cdot c}+\frac{A_\mathrm{p}\tan\frac{\omega\cdot l_\mathrm{p}}{c}+A_\mathrm{c}\tan\frac{\omega\cdot l_\mathrm{c}}{c}}{A_\mathrm{p}-A_\mathrm{c}\tan\frac{\omega\cdot l_\mathrm{p}}{c}\tan\frac{\omega\cdot l_\mathrm{c}}{c}}+3\cdot\tan\frac{\omega\cdot l_\mathrm{p}}{c} \qquad (8),$$

kjer sta: A_c v m^2 – prerez valja, l_c v m – dolžina valja (definirana kot polovica delovnega giba bata).

Z uporabo zapisanih enačb je mogoče izračunati izmere optimalnega resonančnega sesalnega zbiralnika v zelo kratkem času. Žal pa te enačbe ne upoštevajo vseh vplivnih dejavnikov, npr: *spremembo prostornine valja, vpliv krmiljenja ventilov, sprememb v prerezu sesalnih cevi, vpliva hitrosti in tlaka plina na tlačnem valu, vpliv tlačnih nihanj od preostalih valjev pri motorju z več valji, vpliv izpušnega zbiralnika, vpliv segrevanja plina zaradi segrevanja samega sesalnega zbiralnika, vpliv tlačnih uporov v sesalnih ceveh itn.* Poleg tega pa omenjeni avtorji ne podajajo enačb za izračun prostorninskega izkoristka motorja.

where: A_c, m^2 – cross-sectional area of the cylinder; l_c, m – length of the cylinder (set to be equal to half of the piston stroke).

With all these equations it is possible to calculate the dimensions of a tuned or resonant intake manifold, for the defined engine speed, in a short period of time. These equations, however, do not take into account all the relevant factors, such as: *cylinder volume change, influence of valve timing, influence of valve lift, change in pipe cross-sectional area, influence of gas velocity and gas pressure on the wave speed, the influence of waves from other intake pipes in a multi-cylinder engine, the influence of the exhaust manifold, the influence of gas heating, the influence of the friction resistance in the pipes, etc*. Moreover, these equations do not give the value of the volumetric efficiency.

1.2 Enorazsežne numerične simulacije

Z enorazsežnimi numeričnimi simulacijami lahko izračunamo časovni potek tlaka, temperature, masnega pretoka, hitrosti plina itn. v sesalnih ceveh, ki so nadalje namenjeni za izračun prostorninskega izkoristka motorja za izbrane robne pogoje oz. razpored. Optimizacijo sesalnega zbiralnika izvedemo s ponavljanjem izračunov pri različnih robnih pogojih. Osnova za izračune so enačbe enorazsežnega toka neviskoznega plina:

Kontinuitetna enačba:

1.2 One-dimensional simulation calculations

One-dimensional simulation calculations calculate the time trace of pressure, temperature, mass flow, gas velocity, etc. in the intake pipe, and with these results they calculate the volumetric efficiency for a predetermined intake-manifold configuration. The optimisation of the intake manifold can be conducted by analysing these results for several different manifold configurations. The bases for these calculations are the equations of one-dimensional inviscid flow:

Continuity equation:

$$\frac{\partial\rho}{\partial t}+\frac{\partial(\rho\cdot v)}{\partial x}+\frac{\rho\cdot v}{A}\cdot\frac{dA}{dx}=0 \qquad (9)$$

gibalna enačba:

momentum equation:

$$\frac{\partial(\rho\cdot v)}{\partial t}+\frac{\partial(\rho\cdot v^2+p)}{\partial x}+\frac{\rho\cdot v^2}{A}\cdot\frac{dA}{dx}+\rho\cdot v\cdot|v|\cdot f\,\frac{2}{D}=0 \qquad (10)$$

in zakon o ohranitvi energije:

energy equation:

$$\frac{\partial(\rho\cdot e_0)}{\partial t}+\frac{\partial[\rho\cdot v\cdot h_0]}{\partial x}+\frac{\rho\cdot v\cdot h_0}{A}\frac{dA}{dx}+\rho\cdot q=0 \qquad (11).$$

Pri reševanju sistema enačb enačimo stanje plina s stanjem idealnega plina, kar je za potrebe izračunov pri sesalnih zbiralnikih v večini primerov zadovoljivo [6]:

For concluding the equation set, the gas properties are related by an ideal-gas state equation, which is usually sufficiently accurate for engine manifolds [6]:

$$\frac{p}{\rho} = R \cdot T \qquad (12).$$

Enačbe (9), (10) in (11) predstavljajo sistem parcialnih diferencialnih enačb v času t in legi x, ki analitično niso rešljive. Za reševanje se zato uporabljajo numerične metode, ki z napredkom računalništva dajejo vedno natančnejše rešitve.

Equations (9), (10) and (11) represent a system of partial differential equations relating to the time t and the longitudinal coordinate x, and they cannot be solved analytically. For solving these equations, a numerical method must be employed. With the development of computers, more complex methods have been derived, so today it is possible to find very complex and very accurate numerical methods for solving these equations.

2 RAČUNSKI MODEL

Računski model temelji na štirivaljnem štiritaktnem vrstnem motorju z vžigalno svečko, katerega osnovne izmere so podane v preglednici 1. V analizi je uporabljen sesalni zbiralnik s štirimi glavnimi sesalnimi cevmi in štiritočkovnim vbrizgom goriva (sl. 1).

Optimalna vrtilna frekvenca motorja je bila izračunana z enačbami (1) do (8), za nekaj različnih razporedov sesalnega zbiralnika. Za identične razporede so bile izvedene tudi enorazsežne numerične simulacije, na podlagi katerih so bili dobljeni nekateri sklepi. Da bi bili rezultati numeričnih simulacij čim bolj natančni, so bili poleg uporabe predpostavke o idealnem plinu upoštevani naslednji vplivi: *izračun dogajanja v valju, vžig, pretok plina mimo ventilov, pretok plina skozi omejilnike pretoka, izračun dogodkov v zbiralnem prostoru in povezave le tega s sesalnimi cevmi.*

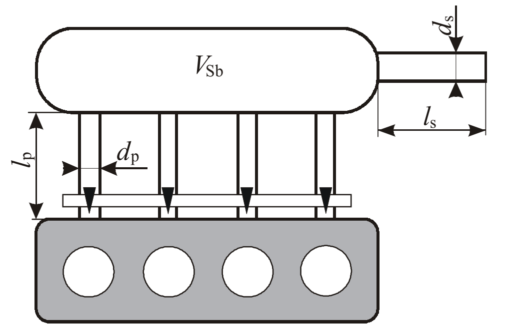
2 CALCULATION MODEL

The calculation model is based on a four-cylinder, four stroke, in-line, spark-ignition engine, whose basic dimensions are shown in Table 1. The intake manifold with four individual primary intake pipes and multipoint injection shown in Fig. 1 is used in the analysis.

The resonant or tuned engine speeds are calculated for several different intake-manifold arrangements, with Eqs.(1) to (8), and the results are shown in the next section (Table 2). For the same intake-manifold arrangements, the one-dimensional simulation calculations are conducted, and with the results from these calculations some conclusions are obtained. In order for the simulation calculations to be as accurate as possible, the calculation model, besides a one-dimensional inviscid flow calculation, comprises the following: *in-cylinder process calculation, combustion, valve flow calculation, calculation of flow through flow restrictions, calculation of processes in plenums and at plenum with pipe connections*. In this manner, a large number of influences is taken into account.

Preglednica 1. *Osnovne izmere motorja*
Table 1. *Basic engine dimensions*

tlačno razmerje ε compression ratio	8,8
premer valja *D* bore D	80 mm
delovni gib bata *H* stroke H	55,5 mm
dolžina ojnice con. rod length L	0,12 m
odprtje sesalnega ventila intake valve opens	40 stopinj pred ZML 40 deg CA before TDC
zaprtje sesalnega ventila intake valve closes	82 stopinj po SML 82 deg CA after BDC
odprtje izpušnega ventila exhaust valve opens	79 stopinj pred SML 79 deg CA before BDC
zaprtje izpušnega ventila exhaust valve closes	30 stopinj po ZML 30 deg CA after TDC



Sl. 1. *Sesalni zbiralnik računskega modela (d_p – premer glavne sesalne cevi, l_p – dolžina glavne sesalne cevi, d_s – premer vstopne sesalne cevi, l_s – dolžina vstopne sesalne cevi, V_{Sb} – prostornina zbirnega prostora zbiralnika)*

Fig. 1. *Intake manifold of calculation model (d_p –primary pipe diameter, l_p – primary pipe length, d_s – secondary pipe diameter, l_s – secondary pipe length, V_{Sb} – plenum volume)*

Znano je, da ima oblika izpušnega zbiralnika velik vpliv na prostorninski izkoristek motorja in da dinamične spremembe tlaka v izpušnem zbiralniku neposredno vplivajo tudi na dinamične spremembe tlaka v sesalnem zbiralniku v času, ko se odprtje sesalnega in izpušnega ventila prekrije. Omenjeni vpliv pri izračunu ni bil upoštevan, kar je bilo zagotovljeno z izbiro robnega pogoja nespremenljivega tlaka takoj za izpušnim ventilom. Na koncu prispevka so za primerjavo podani še rezultati izračuna brez zanemaritve vpliva izpušnega zbiralnika.

It is well known that the exhaust manifold configuration has a significant influence on the volumetric efficiency, and that dynamic changes of pressure in the exhaust pipe affect the dynamic changes of pressure in the intake manifold when there is valve overlapping. The exhaust manifold has not been included in the calculation model in order to neglect its influence. Instead, a boundary condition with constant pressure has been set just behind the exhaust valve. At the end of the paper, for the purpose of comparison, the calculation of the whole model is made.

3 REZULTATI ANALIZ

3 ANALYSIS OF THE RESULTS

Enačbe (1) do (4) neposredno povezujejo optimalno vrtilno frekvenco z izmerami sesalnega zbiralnika n v min^{-1}, medtem ko enačbe (5) do (8) podajajo resonančno frekvenco sesalnega zbiralnika ω_{res} v rad/s in z njo povezano resonančno vrtilno frekvenco motorja n_{res} v min^{-1}:

Eqs.(1) to (4) directly link the dimensions of the intake manifold with the tuned engine speed n (rpm), while with Eqs.(5) to (8) it is possible to calculate the intake manifold's resonant frequency ω_{rez}(rad/s), and with it, it is possible to calculate the resonant engine speed n_{res}:

$$n_{res} = \frac{15 \cdot \omega_{res}}{\pi} \qquad (13).$$

Slika 1 prikazuje najpomembnejše izmere sesalnega zbiralnika. Za vsako od teh izmer so bile izbrane tri različne vrednosti in izračunani optimalna in resonančna vrtilna frekvenca motorja. Pri spreminjanju ene izmere so bile preostale izmere določene kot srednja vrednost izbranih treh vrednosti. Primer: za analizo vpliva dolžine glavnih sesalnih cevi je bila optimalna in resonančna vrtilna frekvenca izračunana za tri različne dolžine glavnih sesalnih cevi l_p=250, 500 in 750 mm, medtem ko so bile preostale izmere nespremenljive: d_p=35 mm, l_s= 200 mm, d_s=50 mm in V_{Sb}=8,5 dm^3. Rezultati izračunov so prikazani v preglednici 2. Razvidno je, da različne enačbe dajo zelo različne rezultate. Na podlagi velikega števila enačb, ki so bile dokazane kot pravilne, velja, da se s povečevanjem dolžine glavnih sesalnih cevi resonančna vrtilna frekvenca motorja zmanjšuje, kar pa ne velja za

Fig.1 shows the most significant dimensions of the model intake manifold. For each dimension three different values were used for calculating the tuned or resonant engine speeds. While changing the value of one dimension, other dimensions are set to the middle of three values. For example, in order to analyse the influence of the length of the primary intake pipe, the tuned and resonant engine speeds are calculated for three different primary pipe lengths l_p=250, 500 and 750 mm, while the other dimensions were as follows: d_p=35 mm, l_s= 200 mm, d_s=50 mm i V_{Sb}=8.5 dm^3. The results of these calculations are shown in Tab.2. From the results it is clear that different equations give significantly different results. According to a large number of equations that have proven to be accurate, by increasing the length of the primary-intake pipe, the resonant engine speed decreases. An exception to this is Eq. 2. The influence of the primary pipe's diameter enačbo (2). Vpliv premera glavne sesalne cevi se tudi razlikuje, saj se po enačbah (2) do (4) in (7) s povečevanjem premera povečuje tudi resonančna vrtilna frekvenca motorja, medtem ko se po enačbah (6) in (8) le ta zmanjšuje. Enačbe (1) in (5) ne upoštevajo premera glavne sesalne cevi. Nadaljnji izračuni kažejo, da veliko enačb ne upošteva premera vstopne sesalne cevi, njene dolžine in prostornine zbirnega prostora pri izračunu resonančne vrtilne frekvence motorja.

Enorazsežne numerične simulacije so bile izvedene s programom AVL Boost. Rezultati izračuna prostorninskega izkoristka motorja pri vseh vrtilnih frekvencah motorja za vse analizirane modele (pregl. 2) so prikazani na sliki 2. Kakor je razvidno iz preglednice 2, imamo sedem različnih razporedov sesalnega zbiralnika, ki so razdeljene s ponavljanjem v pet skupin, vsaka s tremi različnimi razporedi.

Na sliki 2(a) so prikazane krivulje prostorninskega izkoristka za tri različne dolžine glavnih sesalnih cevi. S povečevanjem njihove dolžine se prostorninski izkoristek motorja pri majhnih in srednjih vrtilnih frekvencah poveča, pri višjih vrtilnih frekvencah pa se zmanjša. Pri razporedu sesalnega zbiralnika z najdaljšo cevjo ima prostorninski izkoristek pri vrtilni frekvenci 3600 min^{-1} izrazit vrh, kar lahko razložimo s pojavom resonančnega polnjenja. Primerjava teh rezultatov s tistimi iz preglednice 1 kaže največje ujemanje z enačbo (7).

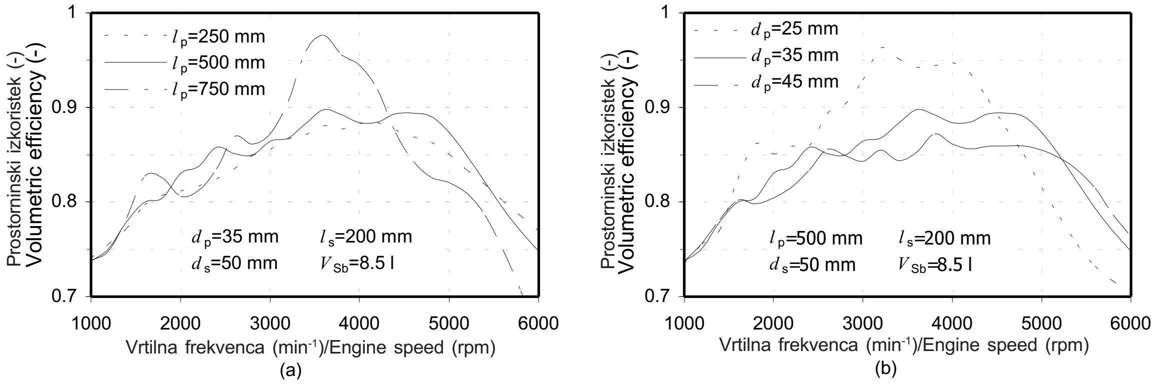
differs according to different equations. Eqs. (2) to (4) and (7) show that when the pipe diameter increases the resonant engine speed also increases, whereas Eqs.(6) and (8) show that when the diameter is increased, the resonant engine speed decreases. Eqs.(1) and (3) to (5) do not take the primary intake pipe's diameter into account. Further calculations show that a large number of the equations shown do not take into account the secondary pipe diameter, the secondary pipe length and the plenum volume when calculating tuned or resonant engine speeds.

One-dimensional simulation calculations were conducted with the AVL Boost program, and for each intake manifold configuration (shown in Table 2), the volumetric efficiency across the whole speed range is calculated. It is clear from Table 2 that there are eleven different intake manifold configurations, which are, with repetition, collected in five groups, each with three different combinations.

Fig.2(a) shows the volumetric efficiency curves for three different lenghts of the primary intake pipe. With an increase in the lenghth of the primary intake pipe, the volumetric efficiency at low and mid engine speed is improved, whereas at high engine speeds the volumetric efficiency is deteriorated. The model with the longest intake pipe has a considerable volumetric efficiency peak at 3600 rpm, which can be considered as resonant charging. The comparison of these results with the results from Tab.1 shows that the best correspondence is obtained with Eq.(7).

Preglednica 2. *Optimalne in resonančne vrtilne frekvence motorja, izračunane z uporabo analitičnih metod*
Table 2. *Tuned and resonant engine speed calculated with analytical expressions*

		Vrtilne frekvence motorja (v min^{-1}), izračunane po enačbi Engine speed (rpm) calculated with equation								
		(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	
l_p mm	250	5120	7210	5890	4180	8220	1670	5600	1650	d_p= 35 mm l_s= 200 mm d_s = 50 mm V_{Sb} = 8,5 dm^3
	500	2870	7580	3300	2340	4600	1580	3650	1560	
	700	1990	7720	2290	1630	3200	1490	2710	1470	
d_p mm	25	2870	4110	3300	2340	4600	1660	3150	1640	l_p= 500 mm l_s= 200 mm d_s = 50 mm V_{Sb} = 8,5 dm^3
	35	2870	7580	3300	2340	4600	1580	3650	1560	
	45	2870	12200	3300	2340	4600	1490	3950	1480	
l_s mm	100	2870	7580	3300	2340	4600	2210	3650	2180	d_p= 35 mm l_p= 500 mm d_s = 50 mm V_{Sb} = 8,5 dm^3
	200	2870	7580	3300	2340	4600	1580	3650	1560	
	500	2870	7580	3300	2340	4600	996	3650	988	
d_s mm	30	2870	7580	3300	2340	4600	957	3650	950	d_p= 35 mm l_p = 500 mm l_s= 200 mm V_{Sb} = 8,5 dm^3
	50	2870	7580	3300	2340	4600	1580	3650	1560	
	70	2870	7580	3300	2340	4600	2170	3650	2140	
V_{Sb} mm	250	2870	7580	3300	2340	4600	1790	3650	1770	d_p= 35 mm l_s= 200 mm d_s = 50 mm l_p = 500 mm
	500	2870	7580	3300	2340	4600	1580	3650	1560	
	700	2870	7580	3300	2340	4600	1430	3650	1420	



Sl. 2. *Vpliv dolžine glavne sesalne cevi (a) in premera (b) na prostorninski izkoristek štirivaljnega motorja*
Fig. 2. *Influence of the primary pipe length (a) and diameter (b) on the volumetric efficiency of a four-cylinder engine*

Na sliki 2(b) so prikazane krivulje prostorninskega izkoristka za tri različne premere glavnih sesalnih cevi. Z zmanjševanjem premera se prostorninski izkoristek motorja pri nižjih vrtilnih frekvencah motorja povečuje, pri višjih vrtilnih frekvencah pa se zmanjšuje. Razvidno je tudi, da se lokalni vrh prostorninskega izkoristka z manjšanjem premera glavnih sesalnih cevi pomika v smeri manjših vrtilnih frekvenc. Primerjava rezultatov z rezultati iz preglednice 1 ponovno kaže najboljše ujemanje z enačbo (7).

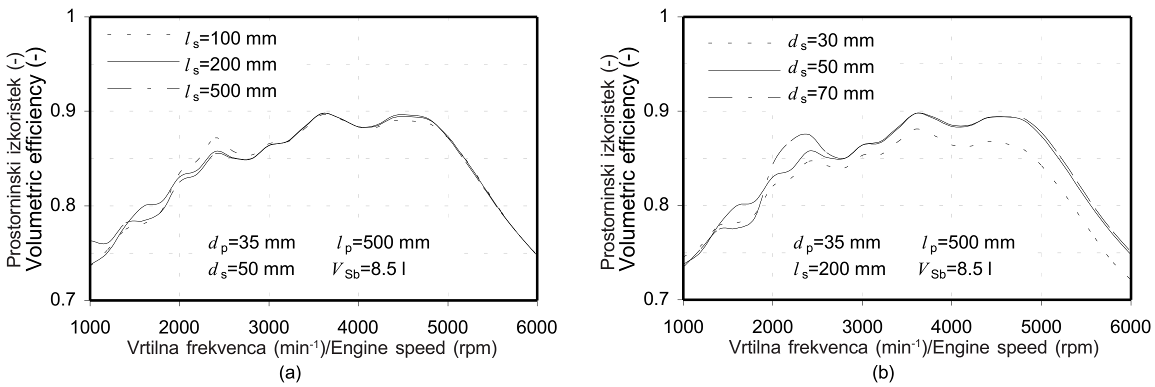
Vpliv dolžine vstopne sesalne cevi na prostorninski izkoristek je razviden iz diagramov na sliki 3a. Pri visokih vrtilnih frekvencah sprememba dolžine vstopne sesalne cevi nima vpliva na vrednost prostorninskega izkoristka, pri nizkih vrtilnih frekvencah pa se kaže v majhnih spremembah strmine krivulje prostorninskega izkoristka motorja. Vpliv dolžine vstopne sesalne cevi je bolj izrazit, če je prostornina zbirnega prostora kolektorja manjša, kar je razvidno s slike 4.

Vpliv premera vstopne sesalne cevi je razviden iz diagramov na sliki 3b. Spreminjanje premera

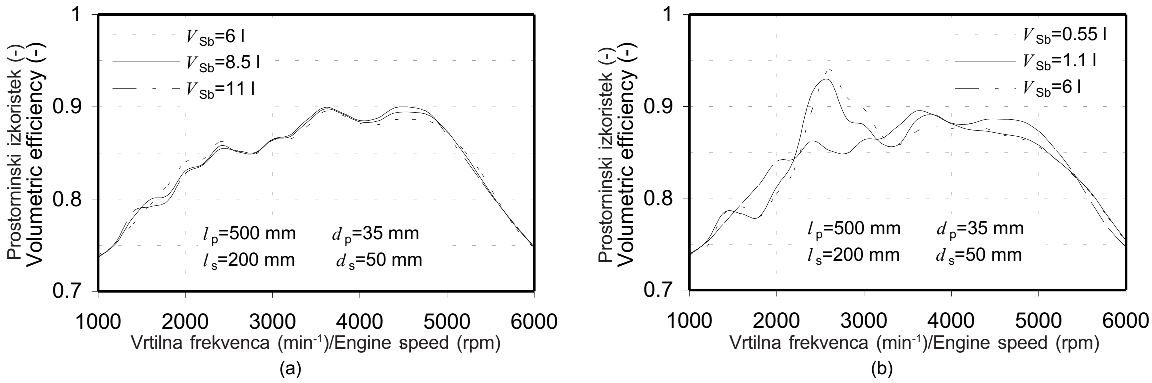
Fig.2(b) shows the volumetric efficiency for three different primary-pipe diameters. With a decrease in the diameter of the primary pipe, the volumetric efficiency at lower engine speeds increases, while at the same time at high engine speeds it decreases. In addition, the local maxima of the volumetric efficiency curve shift to lower engine speeds with a decrease in the diameter of the primary pipe. The comparison of these results with the results from Table1 also shows that the best correspondence is obtained with Eq.(7).

The influence of the length of the secondary intake pipe on the volumetric efficiency is shown in Fig. 3(a). The change in the length of the secondary intake pipe does not cause a change in the volumetric efficiency at high engine speeds, while at low engine speeds it causes very small changes to the shape of the curve. The influence of the secondary intake pipe on the volumetric efficiency would be greater if the plenum volume were smaller (shown in Fig.4).

The influence of the diameter of the secondary intake pipe can be seen in Fig.3(b). The change



Sl. 3. *Vpliv dolžine vstopne sesalne cevi (a) in premera (b) na prostorninski izkoristek štirivaljnega motorja*
Fig. 3. *Influence of the secondary pipe's length (a) and diameter (b) on the volumetric efficiency of a four-cylinder engine*



Sl. 4. *Vpliv prostornine zbiralnega prostora sesalnega zbiralnika na prostorninski izkoristek motorja ((a) – razmeroma velika prostornina; (b) – majhna prostornina)*

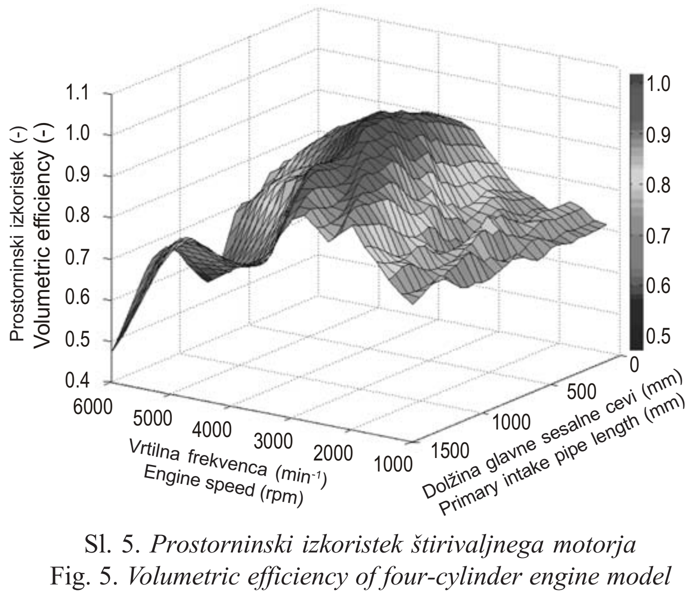
Fig. 4. *Influence of the intake manifold's plenum volume on the volumetric efficiency ((a) – relatively large plenum volume; (b) – lowered plenum volumes)*

vstopne sesalne cevi ne vpliva na prostorninski izkoristek motorja pri visokih vrtilnih frekvencah motorja, pri nižjih vrtilnih frekvencah pa se kaže v spremembi oblike krivulje prostorninskega izkoristka. Če zmanjšamo premer vstopne sesalne cevi preveč, deluje le ta kot dušilo, kar zmanjša prostorninski izkoristek v celotnem območju vrtilnih frekvenc. Zmanjšanje je bolj izrazito z večanjem vrtilnih frekvenc motorja.

Na sliki 4 so prikazani rezultati izračunov za nekaj različnih prostornin zbiralnega prostora sesalnega zbiralnika. Če je prostornina zbiralnega prostora razmeroma velika proti delovni prostornini motorja, majhne spremembe prostornine zbiralnega prostora nimajo vpliva na prostorninski izkoristek motorja (Sl. 4b), toda če prostornino zbiralnega prostora zmanjšamo približno na vrednost delovne prostornine motorja se pojavijo razlike v krivuljah prostorninskega izkoristka motorja. Te razlike so razmeroma majhne, toda v primerjavi z razlikami pri

of the secondary intake pipe's diameter does not change the volumetric efficiency at high engine speeds, but at low engine speeds it changes the shape of the curve. If the secondary intake pipe's diameter is decreased too much, then this pipe becomes the place of choking, and the volumetric efficiency is lowered throughout the whole speed range. At higher engine speeds, the lowering of the volumetric efficiency is greater than at lower engine speeds.

Fig.4 shows the results of simulation calculations for several different intake-manifold plenum volumes. When the plenum volume is relatively big in comparison with the displacement volume of the engine, a small change in the plenum volume will not change the volumetric efficiency curves. (Fig.4(a)). But if the plenum volume is reduced to a value around the displacement volume or smaller, then changes in the volumetric efficiency curve occur. It should be noted that small changes to the small plenum volumes do not result in significant changes in the volumetric efficiency, but in comparison with the results ob-



Sl. 5. *Prostorninski izkoristek štirivaljnega motorja*

Fig. 5. *Volumetric efficiency of four-cylinder engine model*

večji prostornini zbiralnega prostora sesalnega zbiralnika mnogo izrazitejše (sl. 4b). Z manjšanjem prostornine zbiralnega prostora se vpliv elementov pred tem prostorom na prostorninski izkoristek motorja (npr. vhodna sesalna cev) povečuje.

Prikazani diagrami so pridobljeni z numeričnimi simulacijami brez upoštevanja izpušnega zbiralnika. Če bi želeli določiti izmere sesalnega zbiralnika na predstavljeni način, bi bilo treba tudi izpušni zbiralnik vključiti v izračun ter izvesti izračun za celoten motor. Spreminjanje prostorninskega izkoristka v odvisnosti od vrtilne frekvence motorja in dolžine glavne sesalne cevi je razvidno iz diagrama na sliki 5, na podlagi katere lahko dolžino glavne sesalne cevi točno določimo.

4 SKLEPI

Če želimo določiti izmere sesalnega zbiralnika s spremenljivo geometrijsko obliko, je treba natančno določiti izbrane izmere za vsako obratovalno točko. Predstavljene analitične enačbe sicer dajo rezultate hitro, vprašljiva pa je prav njihova točnost. Nobenega dvoma ni, da ne bi trirazsežne numerične simulacije dale mnogo boljši pogled na dogajanje v valju, toda za njihovo uporabo je treba mnogo več vhodnih podatkov. Poleg tega pa je zmogljivost današnjih namiznih računalnikov oz. delovnih postaj še premajhna, da bi bili izračuni opravljeni v sprejemljivem času.

Poiskati je torej treba poravnavo med točnostjo in potrebnim časom izračuna oz. poiskati način, kako optimalno uporabiti vse omenjene metode. V prvi fazi uporabimo za okvirno določitev izmer sesalnega zbiralnika analitične enačbe. Nadaljnjo optimizacijo dosežemo z uporabo enorazsežnih numeričnih simulacij, kar je bilo tudi prikazano in ki v večini primerov dajo dovolj natančne rezultate. Z uporabo trirazsežnih numeričnih simulacij je mogoče pridobiti nekatere stalnice, ki jih nadalje uporabimo v hitrejših enorazsežnih numeričnih simulacijah za doseganje še bolj točnih rezultatov. Na začetku so lahko ti modeli zelo preprosti - z le nekaj elementi motorja, na koncu postopka pa morajo zagotovo vsebovati vse elemente motorja. Na podlagi rezultatov takih analiz je že mogoče točno določiti izmere sesalnega zbiralnika, ki bi dale največjo zmogljivost motorja v dani točki delovanja.

tained with large plenum volumes, the change is considerable (Fig. 4(b)). When lowering the intake-manifold plenum volume, the influence of elements that are located before the plenum on the volumetric efficiency (for instance, the secondary intake pipe) is increasing.

The charts shown are obtained from the simulation calculations of a model without an exhaust manifold. If the dimensions of the intake manifold are to be determined with this kind of simulation calculation, it is necessary to include the exhaust manifold in the model and to conduct the calculations for the whole engine model. The change in the volumetric efficiency caused by a change of one intake-manifold dimension over the whole engine speed range can be shown with a three-dimensional chart (Fig.5), from which this dimension can be precisely selected.

4 CONCLUSION

If an intake manifold with variable geometry is to be designed, then its dimensions must be very precisely determined for every engine working point. The presented analytical equations give results very quickly, but they may not be sufficiently accurate. There is no doubt that three-dimensional simulation calculations would give a much better insight into the charging of the cylinder, but they would require a large amount of input data, and with currently available computers the calculations would last too long.

Therefore, it is necessary to find a compromise between accuracy and the time necessary to obtain the result in the design process, and to find a way how to optimally use all the mentioned methods. In the first phase, the analytical equations will be useful for determining the approximate intake-manifold dimensions. Further optimisation can then be achieved, as has been shown, using one-dimensional simulation calculations, which in most cases give very accurate results. One-dimensional calculations have problems with the flow through sharp bends, pipe junctions and poppet valves. Using three-dimensional calculations on these elements, it is possible to obtain more accurate constants that will be used in simpler and faster one-dimensional models, for even more accurate results. In the beginning these models can be simple, with only a few engine elements, but at the end of the process they must be complete, i.e. they must contain all the engine elements. From the results of these calculations it is possible to select the dimensions of the intake-manifold elements that would give the best engine performance at a defined working point.

5 LITERATURA
5 REFERENCES

[1] Fiala, E., H.P. Willumeit (1967) Schwingungen in Gaswechselleitungen von Kolbenmaschinen. *MTZ* 4(1967) Stuttgart, 144-151.

[2] Broome, D. (1969) Induction ram, part 1, 2 &3. *Automobile Engineer* (1969) London, April issue, 130 – 133, May issue, 180 – 184, June issue, 262-267.

[3] Yagi, S., A. Ishizuya, and I. Fujii (1970) Research and development of high speed high performance, small displacement Honda engines. *SAE paper 700122*.
[4] Ohata, A.,Y. Ishida (1982) Dynamic inlet pressure and volumetric efficiency of four cylinder engine. *SAE paper 820407*.
[5] Ohata, A., H. Saruhashi, I. Matsumoto, Y. Imamura (1985) Acoustic control induction system six cylinder engines. *JSAE rev*. August issue, 8-15.
[6] Winterbone, D.E., R.J. Pearson (2000) Theory of engine manifold design. *Professional Engineering Publishing*, London , ISBN 1 86058 209 5.
[7] Winterbone, D.E., R.J. Pearson (1999) Design techniques for engine manifolds. *Professional Engineering Publishing*, London , ISBN 1 86058 179 X.

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