2008

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# A COMPARATIVE ANALYSIS OF NUMERICAL SIMULATION APPROACHES FOR RECIPROCATING COMPRESSORS

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## ABSTRACT

Simulations of small reciprocating refrigeration compressors have been carried out by using one-, two- and three-dimensional formulations. In all approaches, the valve dynamics was described through a one-degree of freedom model, whereas a finite volume methodology was employed to discretize the governing flow equations. The valve dynamics and the time dependent flow field were coupled and solved simultaneously. Experimental data are used for an assessment of the accuracy of each model. The three-dimensional formulation allowed the inclusion of actual suction and discharge geometries in the simulation and served as the baseline to verify whether simpler and computationally less expensive models can satisfactorily predict the compressor performance. Results included in the analysis are p-V and T-V diagrams, valve displacement and pressure pulsation in the valve suction chamber.

## 1. INTRODUCTION

Several simulation methodologies have been proposed in the literature along the years to predict the performance of reciprocating compressors. The first approaches consisted in quite simple mathematical models with integral formulations for the compression process inside the cylinder and semi-empirical expressions to evaluate the mass flow rate through valves as well as their dynamics. Arguably, the most serious limitation of such models was to assume infinite volumes for suction and discharge chambers, therefore not taking into account the effect of pressure pulsations in the suction and discharge systems on the valve performance. Later, a number of methods were proposed to simulate pressure pulsations, such as acoustic theory (Elson and Soedel, 1974) and non-linear partial differential equations for unsteady one-dimensional (1D) compressible flow (McLaren *et al.*, 1975; Pérez-Segarra *et al.*, 1994; Bassi *et al.*, 2000)

More complex analyses considering two- and three-dimensional (2D and 3D) numerical models started to appear initially for some of the compressor components, such as mufflers (Choi *et al.*, 2000; Fagotti and Possamai, 2000) and valves (Matos *et al.*, 2002). However, three-dimensional numerical simulations for the whole compressor have only recently emerged (Suh *et al.*, 2006; Pereira *et al.*, 2007), as a result of the increasingly availability of computational resources at reduced cost and also because of the development of commercial codes that make it easy to integrate CFD and CAD models. Despite the recent advances in numerical methodologies, the computational cost of a full three-dimensional simulation of a compressor is still impracticable for optimization purposes in which several design options have to be assessed. Therefore, one has still to rely on simpler methodologies that, although requiring some empirical adjustments, can offer satisfactory results for a preliminary design. A good review of the state of the art of CFD models applied to reciprocating compressors was made available by Shiva-Prasad (2004).

In the present work, a comparative analysis of one-, two- and three-dimensional numerical approaches is carried out regarding the simulation of small reciprocating compressors adopted in household refrigeration. In all methodologies, the valve dynamics was described through a one-degree of freedom model, whereas a finite volume methodology was employed to discretize the governing fluid flow equations. The valve dynamics and the time dependent flow field were coupled and solved simultaneously. The three-dimensional formulation, experimentally validated, allowed the inclusion of actual suction and discharge geometries in the simulation and served as the baseline to verify whether simpler and computationally less expensive models can satisfactorily predict the compressor performance. Results included in the analysis are p-V and T-v diagrams, valve displacement, pressure pulsation in the suction and discharge chambers, as well as the coefficient of performance COP.

## 2. MATHEMATICAL MODEL

Valve displacement is modeled by a one-degree of freedom mass-spring model:

$$m_{eq}x + c\,\dot{x} + k\,\ddot{x} = F_p + F_o \tag{1}$$

where $m_{eq}$, $c$ and $k$ are the reed equivalent mass, damping coefficient and stiffness, respectively, which have to be specified. On the other hand, $F_p$ is the flow induced force on the reed and $F_o$ can accommodate any other force, such as reed pre-tension and also stiction that may occur due to the presence of a lubricating oil film between the reed and the valve seat. Finally, $x$, $\dot{x}$ and $\ddot{x}$ are the instantaneous reed lift, velocity and acceleration, respectively. The equivalent mass $m_{eq}$ is determined from data for reed stiffness, $k$, and natural frequency, $f_n$. For the discharge valve, a booster is set to act when the reed valve reaches a certain displacement and a stopper is used to limit the valve lift.

The one-dimensional simulation methodology (Deschamps *et al.*, 2002) solves the one-dimensional transient compressible flow conservation equations for mass, momentum and energy in the suction and discharge systems. Nevertheless, an integral formulation is adopted for the compression process inside the cylinder. Hence, although the mass flow rates through the valves are taken into account in the conservation equations, the flow inside the cylinder is not solved and the cylinder pressure is assumed to be spatially homogeneous throughout the compression cycle. The flow induced force acting on the reed and the mass flow rate through the valve are obtained with reference to effective force area $A_{ef}$ and effective flow area $A_{ee}$, respectively. From the pressure difference across the valve, $\Delta p_v$, $A_{ef}$ is determined from $A_{ef} = F/\Delta p_v$. The effective force area can be understood as a parameter related to how efficiently the pressure difference $\Delta p_v$ opens the valve. On the other hand, for the same pressure drop, $A_{ee}$ expresses the ratio between the actual mass flow rate through the valve and that given by an isentropic flow condition. In the one-dimensional methodology, the viscous effects are modeled with the following procedures: (i) adoption of friction factor coefficients for the flow in the suction and discharge systems, based on classical correlations from the literature; (ii) use of effective area and force areas to characterize the flow through valves. The heat transfer coefficient between the cylinder wall and the vapour was obtained from Annand (1963).

The two- and three-dimensional models were developed with a commercial CFD code, and full details are given in Pereira *et al.* (2007a) e Pereira *et al.* (2007b), respectively. The compressible turbulent flow that prevails in the compressor was solved through the concept of Reynolds-averaged quantities, in which the value of a computed variable represents an ensemble average over many cycles at a specified spatial location. The turbulence transport contribution was modeled through the RNG k-ε model, which has been extensively used and validated for flow through simple geometries of compressor valves (Matos *et al.*, 2002). An ideal gas equation of state completes the system required to solve the compressible flow.

## 3. NUMERICAL SOLUTION PROCEDURE

Figure 1 gives a schematic view of the solution domains and computational grids for each of the numerical simulation methodologies adopted. For the 2D and 3D models a moving grid strategy had to be applied to simulate the compression process due to the presence of moving boundaries, such as the piston and valves. This is not necessary in the 1D model since the compression process is simulated through an integral formulation.

Actual muffler and valve geometries were used in the 3D model. Because an axisymmetric domain was chosen for the 2D formulation, the suction and discharge lines had to be artificially located in opposite sides of the cylinder. On

the other hand, characteristic dimensions for tubes and volumes were defined to model suction and discharge mufflers in the 1D model. As already indicated, the flow restriction imposed by valves is resolved in the 1D model through the use of effective flow and force areas. For the 2D model, the flow through valves is numerically solved, considering circular valves concentrically located with respect to the cylinder axis.



(a) One-dimensional model.



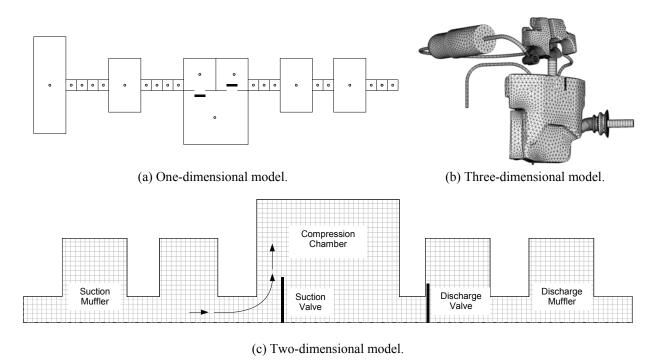(b) Three-dimensional model.



(c) Two-dimensional model.

Figure 1. Schematic view of solution domain and computation grid for each model.

The differential equation for the valve dynamics, Eq. (1), was solved using an explicit Euler method and by considering the force $F_p$ to be constant during the time step. A stiction force, due to the presence of lubricating oil in the valve system, was assumed to act on the reed valves up to a valve lift of 10 μm. The stiction force values prescribed for the suction and discharge valves were 0.5 N and 1.5 N, respectively. The valve parameters and clearance volume were kept the same in all models.

A second-order upwind scheme was adopted to interpolate the flow quantities needed at the control volume faces, whereas a first-order upwind scheme was selected for the 1D model. The coupling between the pressure and velocity fields was achieved with the SIMPLEC scheme. The system of algebraic equations was solved with a segregated implicit algorithm.

In the 1D model, suction and discharge systems are coupled with the cylinder through estimates for valve mass flow rates, obtained from predictions of pressure difference between the cylinder and suction/discharge chambers. Therefore, for each time step mass flow rates are used as boundary conditions for the suction and discharge systems, being equal to zero when the valve is closed. With a new solution for the pressure in the suction and discharge chambers, new values for the valve lift and mass flow rate can be evaluated and transferred as an input for the cylinder integral solution domain, allowing the compression process to be calculated for the next time step. Then, the pressure in the cylinder allows the valve mass flow rate to be calculated and used as boundary condition for the suction and discharge systems. This procedure is repeated for a number of compression cycles until convergence is found.

Evaporation and condensation pressures were imposed as boundary conditions at the extremities of the suction and discharge mufflers, respectively. The muffler solid walls were assumed to be adiabatic because no data was available for the external heat transfer condition. For this reason, the inlet temperature in the suction muffler was set

to the value experimentally measured in the suction chamber, so as to have the correct temperature for the gas entering the cylinder.

For the 2D and 3D models, all velocity components were set to zero at the solid walls, but for the reed and piston surfaces the velocities were obtained from Eq. (1) and the crankshaft mechanism, respectively. A turbulence intensity of 3% and turbulence length scale corresponding to the tube hydraulic diameter were adopted as the inlet boundary condition for the turbulent flow in the mufflers. The evaporation pressure and suction line temperature were employed as initial estimates for the suction muffler and for the compression chamber. For the discharge muffler, the condensation pressure and discharge line temperature were chosen instead.

The convergence of the iterative procedure is assessed by examining whether the compressor operation conditions are cyclically repeated. In the 2D and 3D models, due to numerical stability aspects of the solution procedure and also because of constraints associated with the deforming mesh method, two different time steps had to be used according to whether the valves were open or closed. Accordingly, the time steps were set to values corresponding to 0.5 degrees of crankshaft angle when the valves were closed and 0.1 otherwise. In the 2D and 3D models, 3 cycles were required to establish a periodic condition, with a processing time of approximately 2 and 20 hours per cycle, respectively, on a computer with a single Pentium IV 3 GHz processor. On the other hand, the 1D model used a constant time step corresponding to 0.05 degrees of crankshaft angle, taking approximately 30 seconds to perform 10 cycles and achieve a periodic condition.

## 4. RESULTS AND DISCUSSIONS

The 3D model is expected to be the most accurate methodology because it can better characterize the compressor geometry. For this reason, the results obtained with the 3D model were initially compared with experimental data for pressure and temperature in the suction chamber, pressure and temperature in the cylinder and valve displacement as a function of crankshaft angle. All measurements were carried out in a calorimeter facility for two 60 Hz compressors operating with R134a and with cooling capacities of 170 and 270 W.

Figure 2 shows a comparison between numerical and experimental results for the compressor p-V diagram, corresponding to the suction and discharge processes. In this figure, the pressure value was normalized by the pressure condition in the suction line or discharge line, depending on the process being examined. It should also be noticed that the instantaneous compression chamber volume was normalized by the compressor total volume displacement. For both compressors analyzed in different operating conditions the results are similar. The agreement of the numerical results with the experimental data is very good in the case of the suction process, but some discrepancies are observed for the discharge process. The first point to be noticed is that the p-V diagram predicted during the discharge process is wider than that obtained experimentally. A possible reason for this could be some discrepancy between the actual and modeled clearance volumes and, also, the failure of the ideal gas equation of state to describe the compression process. Since the compressibility factor is equal to unity in the ideal gas model, for a certain volume the numerical result for pressure is expected to be higher than the value a real gas hypothesis would predict. The second interesting aspect is the higher pressure pulsation amplitude verified in the experimental data during the discharge process. This behavior could be associated with the pressure tap geometry in the cylinder wall. Pressure pulsations in the suction chamber are depicted in Fig. 3 for two system conditions, represented by condensation temperatures $T_c$ = 54.4 and 40.5 ºC. The agreement between numerical and experimental data is very good, concerning both phase and amplitude. The pressure pulsation in the suction chamber affects the valve dynamics and mass flow rate and, hence, its correct prediction is of primary importance in the compressor simulation.

A further important part of the numerical procedure validation was to verify the capability of the model to predict variations in the compressor performance due to modifications in its design and operation condition. In this respect, Table 1 presents results for variations in power consumption after changing the condensation temperature and the diameter of discharge valve orifice. In the former case, condensation temperature was decreased from 54.4ºC to 40.5ºC. In the latter, the diameter of the discharge orifice was increased by 30%. As can be seen, the numerical model was able to predict the relative variations for both test cases in close agreement with the measurements. In spite of some discrepancies between numerical and experimental results for the pressure inside the cylinder during

the discharge process (Figure 2b), there is a clear evidence that the numerical model can be used to predict power consumption variations originated by modifications in compressor design and operation condition.
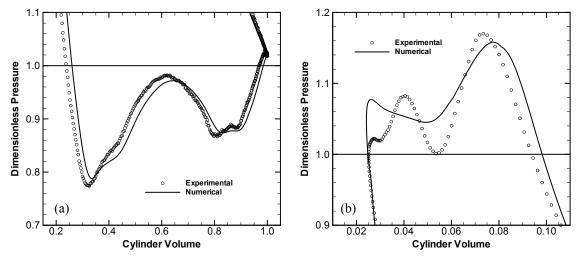


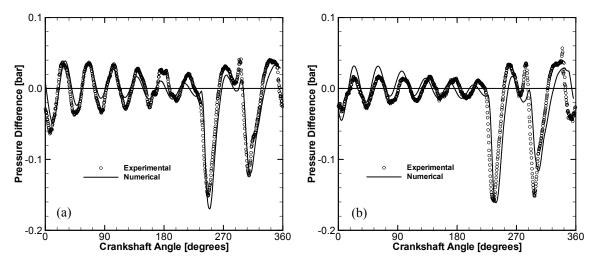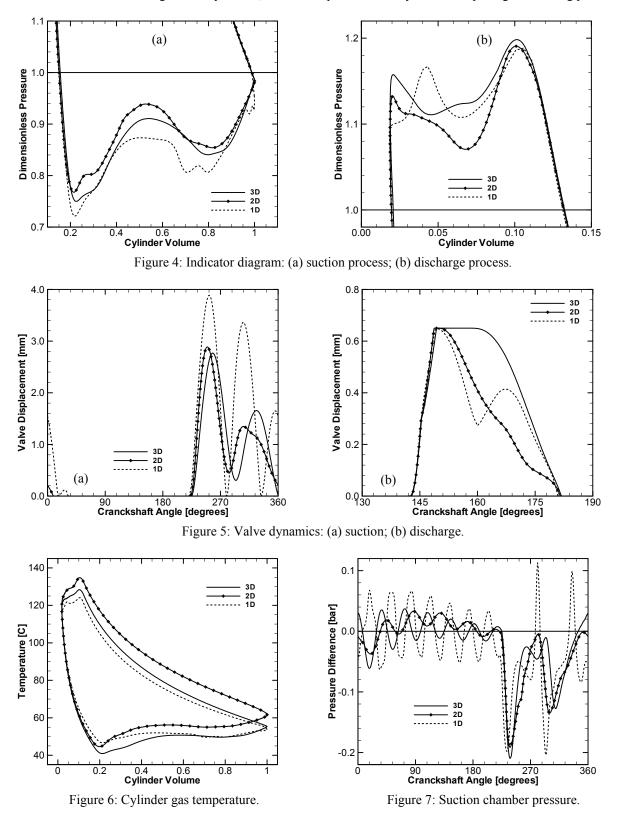Figure 2: Indicator diagram: (a) suction process; (b) discharge process.



Figure 3: Suction chamber pressure: (a) $T_c = 54.4°C$; (b) $T_c = 40.5 °C$.

Table 1: Numerical and experimental results for variation of power consumption.

| Modification | Indicated power | | Suction power | | Discharge power | |
|---|---|---|---|---|---|---|
| | Exp. | Num. | Exp. | Num. | Exp. | Num. |
| Operation condition | -11.5% | -7.5% | +8.3% | +8.3% | +38.1% | +37.0% |
| Discharge orifice | -3.3% | -2.8% | -3.6% | +1.8% | -32.5% | -33.0% |

The previous test situations used to validate the 3D model were considered also for a comparative analysis between the 1D, 2D and 3D models, considering the compressor with higher cooling capacity (270 W). Figures 4, 5, 6 and 7 show a comparison between numerical predictions of the three models for p-V diagram, valve dynamics, T-V diagram and pressure in the suction chamber, respectively. The 1D model predicted that the opening of the suction valve occurs at approximately 4 degrees earlier than estimated with the 2d and 3D models (Fig. 5a). In addition to that, the dynamics of the suction valve obtained with the 1D model is quite different in comparison with the results of the two other models, with higher pressure pulsation amplitude in the suction chamber. By examining some

possible reasons for such a difference, it was found that the valve dynamics evaluated in the 1D model is much more affected by the damping coefficient *c* than in the other models. On the other hand, despite some visible differences in the results for the discharge valve dynamics, the models predict virtually the same opening and closing positions.

Figure 4: Indicator diagram: (a) suction process; (b) discharge process.

Figure 5: Valve dynamics: (a) suction; (b) discharge.

Figure 6: Cylinder gas temperature.

Figure 7: Suction chamber pressure.

As shown in Fig. 6, the T-V diagrams predicted by the 1D and 3D models are in close agreement, which could be an indication that the correlation of Annand (1963) adopted in the 1D characterizes the heat transfer between the vapor and the cylinder walls more accurately than does the 2D model. This is a consequence of the fact that the adoption of an axisymmetric geometry in the 2D model gives rise to a radially distributed flow at the exit of the suction valve, impinging the vapor against the cylinder walls. In the actual suction valve geometry, considered in the 3D model, a much smaller amount of gas is directed towards the cylinder wall, reducing the superheating effect.

Tables 2 and 3 compare variations in power consumption, cooling capacity and COPpV estimated by the three models as a response to changes in the condensation temperature and discharge orifice diameter. In all simulations, the greatest difference between results for cooling capacities was about 3%. Despite the 1D model overestimated the variation in the discharge power, satisfactory trends were observed for all the remaining quantities. Taking into account the simplifications adopted in the 1D and 2D models, and their computational cost, it can be concluded that their use is a good compromise for numerical simulations of reciprocating.

Finally, Figs. 8 and 9 compare the results for valve dynamics and pressure pulsation in the suction chamber, obtained with the 1D and 3D models. For the 1D model, it was also tested the influence of the damping coefficient, $c$, on the results, by increasing its original value four times. It is clear that when the damping coefficient is increased, there is a closer agreement between the results of the 1D and 3D models. The necessity of a higher damping coefficient in the 1D model may be an artificial form to compensate errors due to the absence of flow transit effects in the effective flow and force areas. The valve dynamics has a strong effect on the amplitude and phase of the pressure pulsation in the suction chamber. As indicated in Figs. 8 and 9, the 1D model can predict results for pressure pulsation in agreement with those of the 3D model when the prediction for the valve dynamics is improved.
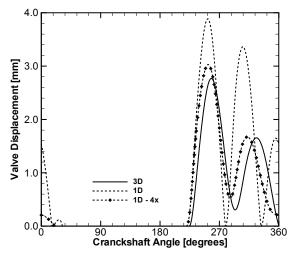
## 5. CONCLUSIONS

A comparative analysis of three simulations methodologies, represented by 1D, 2D and 3D differential formulations, was carried out regarding the simulation of small reciprocating refrigeration compressors. In all methodologies, the valve dynamics was described through a one-degree of freedom model, whereas a finite volume methodology was employed to discretize the governing fluid flow equations. Results included in the analysis are p-V and T-v diagrams, valve displacement, pressure pulsation in the suction and discharge chambers, as well as the coefficient of performance COP. Some differences observed between the numerical results are attributed to the use of effective flow and force areas in the 1D model and by the adoption of an axisymmetric geometry in the 2D model. Nevertheless, all models predicted very similar variations in the compressor performance due to changes in the design conditions, such as the condensing temperature and valve orifice diameter. It is concluded that the simpler, and much less computationally expensive, models are adequate for preliminary analyses of the compressor design.

Table 2: Variations due to condensation temperature change.

| Simulation Model | Indicated Power | Suction Power | Discharge Power | Cooling Capacity | COPpV |
|---|---|---|---|---|---|
| 3D | -8,5% | +5,4% | +33,7% | +6,1% | +16,0% |
| 2D | -8,6% | +6,0% | +34,9% | +6,9% | +16,9% |
| 1D | -7,6% | +8,9% | +51,8% | +9,2% | +18,2% |

Table 3: Variations due to discharge orifice increase

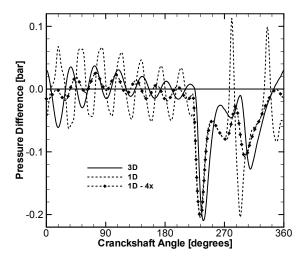| Simulation Model | Indicated Power | Suction Power | Discharge Power | Cooling Capacity | COPpV |
|---|---|---|---|---|---|
| 3D | -2,1% | -0,6% | -32,3% | -0,3% | +1,8% |
| 2D | -3,4% | -1,3% | -42,9% | -0,9% | +2,6% |
| 1D | -1,9% | -0,3% | -50,3% | -0,2% | +1,7% |

Figure 8: Suction valve dynamics.



Figure 9: Suction chamber pressure.

## REFERENCES

Annand, W.D., 1963, Heat Transfer in the Cylinders of Reciprocating Internal Combustion Engines, *Proc. IMechE*, v. 177, pp. 973-996.

Bassi, F., Pelagalli, L., Rebay, S., Betto, A., Orefice, M., Pinto, A., 2000, Numerical Simulation of a Reciprocating Compressor for Household Refrigerators, *Proc. Int. Compressor Engrg. Conf. at Purdue*, pp. 97-104.

Choi, J.K., Joo, J.M., Oh, S.K., Park, S.W., 2000, Smart Suction Muffler Design for a Reciprocating Compressor, *Proc. Int. Compressor Engrg. Conf. at Purdue*, pp. 619-626.

Elson, J.P. and Soedel, W., 1974, Simulation of the Interaction of Compressor Valves with Acoustic Back Pressures in Long Discharge Lines, *J. Sound and Vibration*, v.34, n.2, pp. 211-220.

Fagotti, F., Possamai, F.C., 2000, Using Computational Fluid Dynamics as a Compressor Design Tool, *Proc. Int. Compressor Engrg. Conf. at Purdue*, pp. 137-144.

McLaren, J.F.T., Tramschek, A.B., Sanjines, A. and Pastrana, O.F., 1975, A Comparison of Numerical Solutions of the Unsteady Flow Equations Applied to Reciprocating Compressor Systems, *J. Mech. Engrg Science*, v.17, n.5, pp. 271-279.

Pérez-Segarra, C.D., Escanes, F. and Oliva, A., 1994, Numerical Study of the Thermal and Fluid-Dynamic Behaviour of Reciprocating Compressors, *Proc. Int. Compressor Engrg. Conf. at Purdue*, pp. 145-150.

Deschamps, C.J., Possamai, F.C., Pereira, E.L.L., 2002, "Numerical simulation of pulsating flow in suctions mufflers", *Proc. Int. Compressor Engrg. Conf. at Purdue*, Paper C11-4.

Matos, F.F.S., Prata, A.T. and Deschamps, C.J., 2002, "Numerical simulation of the dynamics of reed type valves", *Proc. International Compressor Engineering Conference at Purdue*, West Lafayette, USA, C15-2.

Pereira, E.L.L, Deschamps, C.J., Ribas, F.A., 2007a, "Numerical modeling of a refrigeration reciprocating compressor", *Proc. 19th Int. Congress of Mechanical Engineering*, Brasília, Brazil, Paper 1677.

Pereira, E.L.L, Deschamps, C.J., Ribas, F.A., 2007b, Performance Analysis of Reciprocating Compressors through CFD Simulation, *Proc. IMechE Int. Conf. Compressors and Their Systems*, London, pp. 309-318.

Shiva Prasad, B.G., 2004, CFD for Positive Displacement Compressors, *Proc. Int. Compressor Engrg. Conf. at Purdue*, 7 p.

Suh, K., Heo, D.N., Kim, H.S., 2006, CAE/CFD Application for Linear Compressor, *Proc. Int. Compressor Engrg. Conf. at Purdue*, 8 p.

## ACKNOWLEDGEMENTS